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WORKSHOP PRACTICE

VOLUME VII

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PLAIN BEARINGS

BY

A. J. AIERS

BALL AND ROLLER BEARINGS

BY

A. W. JUDGE, Wh.Sc., A.R.C.S., A.M.I.A.E.

PETROL ENGINE FITTING AND ASSEMBLY

BY

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AND

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REVISED BY

W. J. PECK, M.Eng., A.M.I.Mech.E , A M.I.A.E.

WORKSHOP PRACTICE

A PRACTICAL WORK FOR THE DRAUGHTSMAN,
THE MECHANIC, THE PATTERN-MAKER, AND
THE FOUNDRYMAN

SECOND EDITION

EDITED BY
WILLIAM H. ATHERTON
M.Sc., M.I.Mech.E.

VOLUME VII

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PREFACE

THIS and the subsequent volume have been compiled expressly for the benefit of those mechanics whose duty it is to attend to the fitting, erection, and testing of internal combustion engines, steam engines, and steam turbines. The present volume deals with petrol engine and gas engine fitting and assembly. Preliminary to considering actual fitting and erection operations it has been deemed advisable to offer to the reader some useful matter on the subject of bearings which, of course, are of primary importance in connection with all engine work.

In Section XXIX he will find a very complete treatment of plain bearings in which every conceivable type is touched on. Bearings for marine purposes, automobile work, stationary engines, locomotives, and the numerous other purposes are all dealt with. The author also deals with oil-less bushes, such as are used in watches, magnetos, and other delicate precision instruments. Bearing construction, mounting and fitting are all clearly described, and the lubrication of each type receives the thorough treatment which is deserved. Various types of oil pumps and indicators are discussed, and the recognized systems of oil grooves explained.

Ball and roller bearings are the subject-matter of Section XXX. These bearings, which are used where loading and speeds are high and where the elimination of friction is of paramount importance, are very fully described and illustrated. The fundamental difference between plain bearings and ball or roller bearings, i.e. the substitution of rolling friction for sliding friction, forms the essence of the author's introductory remarks.

He thereafter proceeds to describe the principles, construction, and mounting of the Hoffmann, Skefko, Hyatt, and Timken bearings, and to detail their various applications.

The two final sections in this volume concern the fitting and erection of internal combustion engines—petrol and gas engines. These sections are very well illustrated by clear sketches and diagrams.

Valve grinding, valve and ignition timing, piston ring lapping, bearings and their alignment, working clearances, joint making, and lubrication are all in turn discussed, and emphasis laid upon the chief things to be noticed and attended to by the mechanic during the course of erection. The matter of engine foundations is also dealt with briefly.

In the petrol engine section, the four-stroke engine alone is considered. This is excusable, since little difference exists between the fitting and assembly of the two types, except that in the case of the two-stroke the operations are rendered vastly more simple due to the entire absence of complicated valve gear. The two-stroke engine, it should be remembered, is used mainly for light-weight motor-cycles and for small stationary purposes.

The testing of internal combustion engines is dealt with in Volume VIII.

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PLAIN BEARINGS

BY

A. J. AIERS

SECTION XXIX

PLAIN BEARINGS: TYPES AND FITTING

INTRODUCTION

BEARINGS are important parts of any machine, as they act as contacts between the main frame and all the moving parts. There are many bearings on each machine, and if one fails under service, the whole machine must stop. Bearings are also the parts of the machine which bear the friction, and a designer aims to cut this down to a minimum.

Friction tends to stop the moving parts and generates heat. Despite the presence of a lubricant, which acts as a layer between the surfaces and stops them from seizing, there is still a certain amount of friction. If the heat generated be excessive, it must be carried away by the lubricant and then the lubricant cooled. Not only must a bearing run cool, but it must not wear rapidly, and this means a low bearing pressure.

Of the many types of bearings, that most used is the journal bearing, Fig. 1, in which the load acts at right angles to the axis of the shaft.

THRUST BEARINGS

In thrust bearings the load acts parallel to the axis, as illustrated in Fig. 2, where it is used in conjunction with a journal bearing. The two steel washers take care of the end thrust. If a spindle has to carry a load and also take a thrust, a conical bearing (Fig. 3) could be used which would take care of both loads, but the friction would be excessive, and usually separate journals and thrust bearings are fitted. When the shaft or spindle is in the vertical position, as in Fig. 4, the bearing is given the name of footstep bearing. This is fitted with single or multiple thrust washers, according to the load to be carried. Floating bushes are fitted where the housing and shaft are of suitable material.

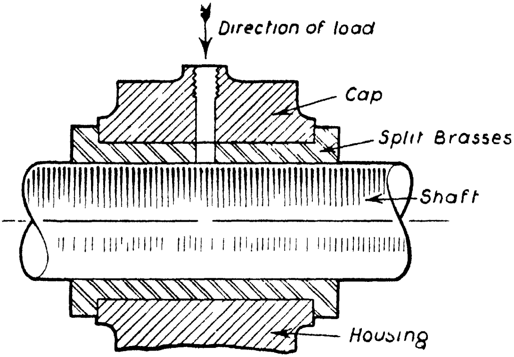


FIG. 1. SECTIONAL DRAWING OF JOURNAL BEARING

In these bushes, the fits of both the outside and the bore allow running clearance, but end pieces must be fitted to locate the bush.

On most machines it is necessary to transmit motion to slides, and this is usually accomplished by a screw and nut mechanism. In this case the nut is an important bearing, and one which is often overlooked. The usual material for nuts is bronze, but some are made by casting a whitemetal nut, using the screw as a mandrel.

SOLID AND ADJUSTABLE BEARINGS

Bearings are divided into solid and adjustable types. A solid type is satisfactory as long as slight play does not affect the use of the machine, but when wear takes place, a new bearing is necessary. With adjustable bearings, if play develops, it can soon be taken up by removing metal from the faces of the split bush or steps. Special bearings are also made in which there are four pieces, namely, top and bottom and side pieces, which can be adjusted by wedges.

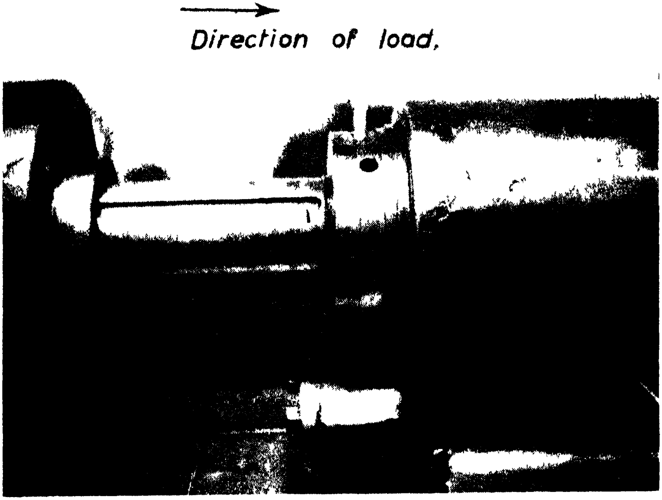


FIG. 2. A TYPICAL THRUST BEARING

Another type of adjustable bearing is the tapered bush, and also the split bush. A tapered bush is illustrated in Fig. 5, and the adjustment is made by moving the bush endways, by means of screwed collars. Being split, the housing compresses the bush as it is forced farther in.

LATHE BEARINGS

With lathes and other machine tools, the main bearings must be a good fit, and must also be capable of being kept in this condition, otherwise the play in the spindle would not allow accurate work to be produced. As will be seen from Fig. 6, the main bearings are split

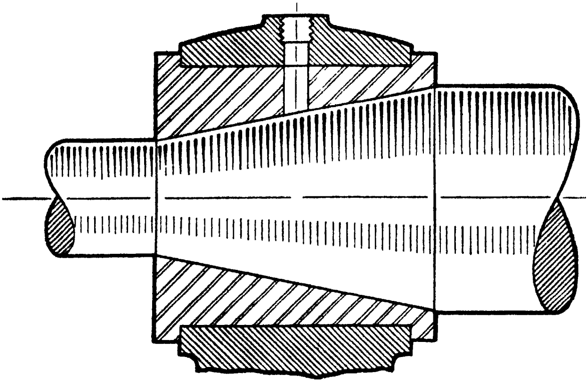


Fig. 3. Conical Bearing

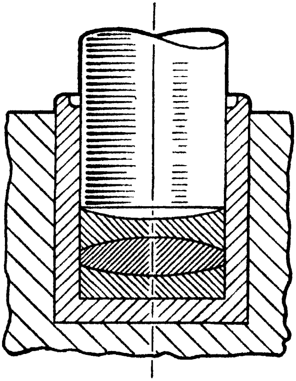


Fig. 4. Section of Footstep Bearing

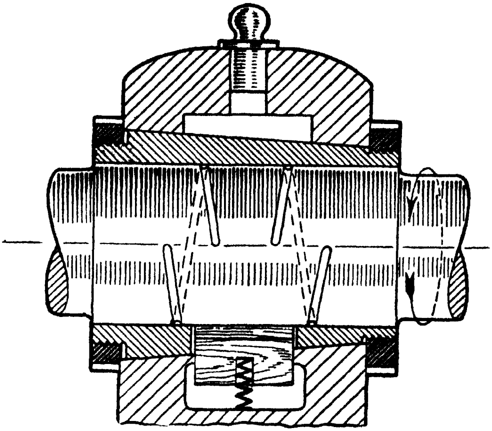


FIG. 5. ADJUSTMENT ON TAPERED BUSH BEARING

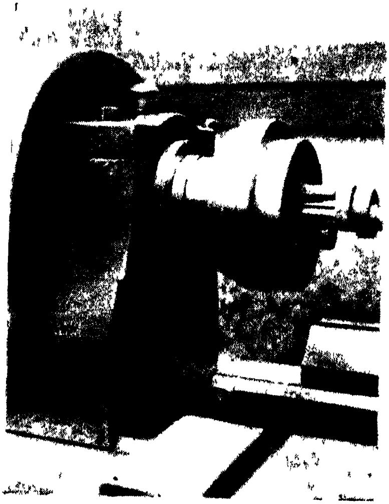


FIG. 6. BEARINGS OF LATHE HEADSTOCK

so that play can be taken up. The bottom half of the brass rests in the headstock and is, of course, pegged to prevent rotation, while the top half is held down by a cap through which the lubricant is supplied. Another headstock bearing is shown in Fig. 7, the bearing material in this case being whitemetal, and the thrust is taken on a thrust race fitted at the rear bearing The ways and beds of machine tools are bearings where only sliding action takes place, but, nevertheless, they require oil grooves to distribute the oil, and also need provision to keep out cuttings which would spoil the surface.

FIG. 7. HEADSTOCK BEARING DISMANTLED

Some of these slides are extensive, as here the object is not to keep down friction so much as freedom from wear, and also stiffness to withstand the heavy cuts.

SHAFTING BEARINGS

Although line shafting bearings now consist of either the ball or roller type, there is hardly a factory in which some of the shafting and most of the countershafts are not of the plain type. With shafting it is important that the various bearings be kept in line, and, to facilitate this, they are adjustable in the hangers, either by screwed pins or other kinds of spherical mounting.

The methods of lubrication are usually by means of ring oilers, in which a slot is cut in the top of the bearing, and a loose ring slipped into this slot. As the shaft revolves it turns the ring, the lower half of which dips in an oil bath and carries the oil up to the surface of the bearing.

FIG. 8. ADJUSTABLE HANGER

In another type, instead of a loose ring, a collar is joined on the shaft which lifts the oil in a similar manner. This, of course, is more certain than a ring, and will operate at any speed. Ring oiling is not satisfactory at very high speeds as slip occurs. When once the reservoirs are filled, the bearings will run for months without much attention. The hanger in Fig. 8 has an oil level in the form of a glass tube fitted, so that a correct supply is always available. The bearing is adjustable for height by means of the top and bottom support arms, which are screwed and fitted with a locking screw. The bearing can also pivot around the supports so that alignment is fairly easy.

On shafts of large diameter, small separate pumps are used, driven from the shafting to supply the lubricant and ensure a perfect supply. In other special cases, the bearing itself is water-cooled by means of inserted pipes through which a flow of water is maintained.

The details of standard plummer blocks (Fig. 9) are sometimes similar to hangers and the brasses are pegged in, or in older types the outside section of the brasses are square, hexagon, or octagon, and fit into correspondingly shaped housings. Where whitemetal bearings are used they should be held securely in the housings, either by pegging, grooving, or serrating.

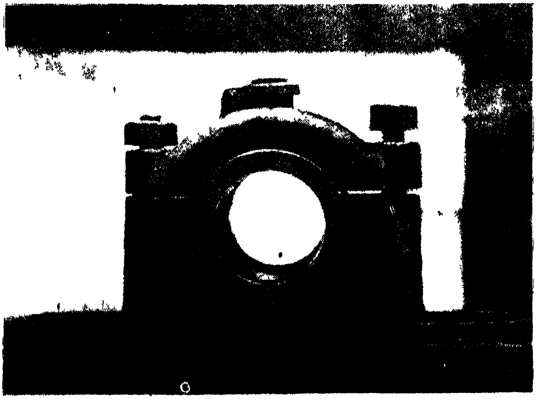


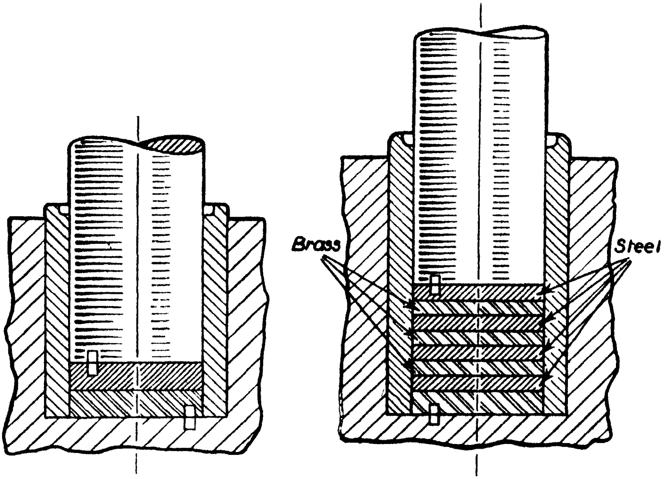
FIG. 9. SIMPLE PLUMMER BLOCK, HAND OILED

FOOTSTEP BEARINGS

With shafting and hangers, there is hardly any thrust to trouble about, but if the shaft is vertical, there is the weight of the shafting and the pulleys to support. In the simpler type of footstep bearing (Fig. 10), a pair of thrust washers are used, one fixed to the end of the shaft and the other fixed in the bearing housing. In a more important case, several discs are used, and the discs are alternately hardened steel and bronze, the end ones being fixed as before.

COLLAR TYPE BEARINGS

With a shaft of large diameter, a step bearing is unsatisfactory, as the velocities and, therefore, the



FIGS. 10 AND 11. SINGLE AND MULTI-WASHER FOOTSTEP BEARINGS

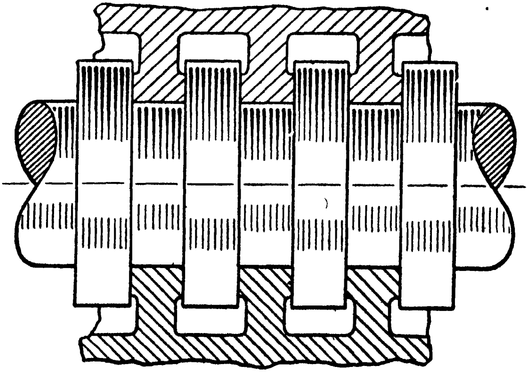


FIG. 12. COLLAR TYPE BEARING

wear vary at different points. For instance, imagine a shaft nearly 12 in. diameter, the velocity, say, at 200 rev. per min. would vary from nil at the centre of the shaft to 300 ft. per min. half way, and 600 ft. per min. at the outside diameter. The larger part of the centre is therefore cut away and only an outer ring used, but a much better way is to use a collar type bearing. As shown diagrammatically, several collars are turned on the shaft, and these bear up against similar pieces in the box, thus increasing the bearing surface.

Vertical bearings such as used on water turbines, pumps, and vertical generators are the most difficult to lubricate, as the oil runs away by gravitation. An oil groove should be cut either right or left hand according to the rotation of the shaft, which when revolved will act on the principle of an archimedean screw, and carry the oil up the shaft to the bearing.

MACHINE TOOL BEARINGS

On precision machines, and especially grinding machines, great care must be taken to ensure that no dust or fine abrasive can obtain access to the bearings, as otherwise wear and the consequent bad finish and chatter marks will be produced. With internal grinding machines, the spindles also have to run at extremely high speed, and therefore need careful adjustment. The bearings are usually tapered, and also screwed each end for adjusting rings. The taper is used for adjusting the size of the hole, as the bearing is forced in and out of the housing by means of the screwed rings at either end. A complete dismantled spindle is shown in Fig. 13. As will be seen, there is a taper at both ends of the spindle, and a pair of adjustable bearings are fitted. These are screwed into the housing until all end-play is removed, and only running clearance allowed. With taper bearings as described, the thrust is not taken by the taper, which would cause it to bind, but by a separate thrust washer; the taper is only for adjusting running clearance, instead of having to strip down and file faces as in the standard two-piece type of bearing. On the drilling machine spindle (Fig. 14), which only has vertical movement, a plain bearing is used, but is split so that all play can be taken up.

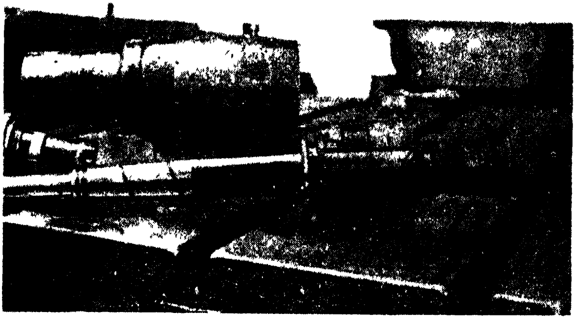


FIG. 13. SPINDLE WITH TWO ADJUSTABLE TAPER BEARINGS

Although bearings are usually split across the centre, this is often only because of economical production. The splitting should be at an angle or at the point of least pressure as indicated in Fig. 15. It is not always necessary to have a complete bearing. For instance, in railway wagon axles (Fig. 16) and also such household things as mangles, only half bearings are used as the pressure is only one way, and the other half if fitted would only be waste metal.

The bearing only fits on the top half, and the bottom half has an oiling pad held in contact by a spring. Knife edges are another type of bearing, although in an entirely different class. They are much used on weighing and balancing machines, where the least possible amount of friction can be allowed.

AUTOMOBILE BEARINGS

In some motor-cars we have the main engine bearings of the journal type, also gudgeon pin bearings which

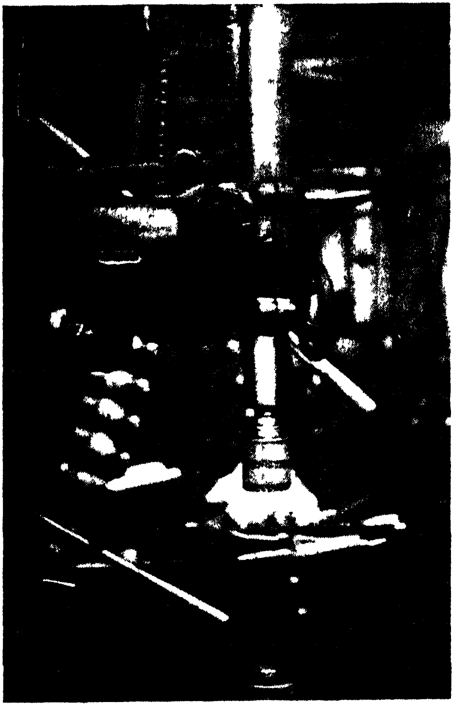


FIG. 14. DRILLING MACHINE WITH SPLIT BEARING

only have intermittent action. The materials vary, whitemetal being used for main bearings, phosphor bronze for gudgeon pins, oil-less bushes for shackles, wood bushes for shock absorbers, and fibre bushes for contact-breaker pivots. Where a small amount of wear only takes place, solid one-piece bushes are used, but often these have to be split so that the parts can be assembled. The big end and main crank journals are mostly of whitemetal in a gunmetal shell. Figs. 17 and 18 illustrate a set of main bearings and a connecting rod with a set of bearings and the packing shim. In all cases the oil hole feed is in the top half, but it will be noticed that the oil distributing grooves are in the top half of main bearings, and in the bottom half of the connecting rod bearings. This is in accordance with the idea that oil feeding grooves should be

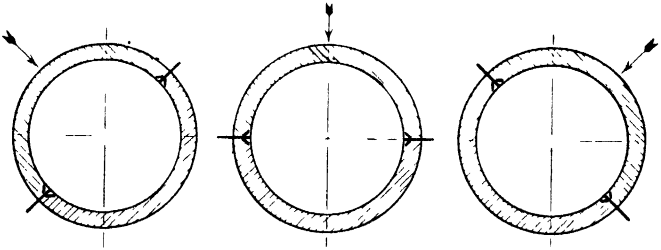


FIG. 15. DIAGRAM OF SPLIT BEARINGS

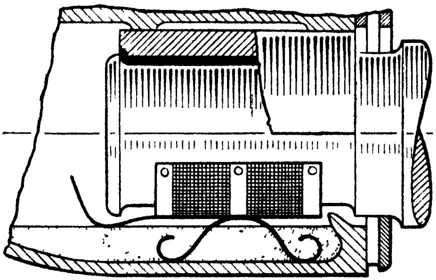


FIG. 16. DRAWING OF AXLE BOX AND BEARING

at point of least pressure. When the engine is driving the car, the load is delivered by the connecting rod being forced down, and this in turn transmits the power to the crank; therefore the pressure is on the top half of the rod and the bottom half of the main bearings.

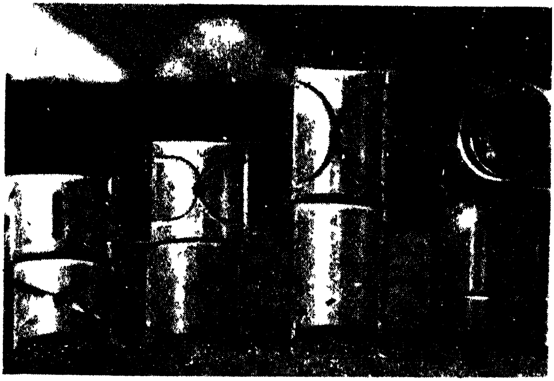


FIG. 17 WHITEMETAL LINED BEARINGS FOR PETROL ENGINE

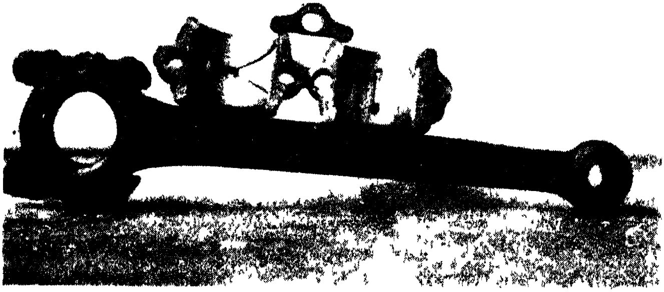


FIG. 18. CONNECTING ROD ASSEMBLED AND SET OF BEARINGS

The gudgeon pin bearing in some engines is force fed, but many rely upon splash feed, or use a scraper ring, which removes surplus oil from the cylinder wall and delivers it to the bearing.

BALANCE OF MOVING PARTS

As engines now run at very high speeds, it is necessary to balance both the rotating parts and also the reciprocating parts, so as to ease the work of the bearings and to produce a smoother running engine. The small end of the connecting rod is finished to size, and also the oil grooves cut by broaching. When broaching the hole, the last few teeth on the tool do not cut but only burnish. They are of round section and slightly compress the metal, giving it a hard surface which will resist wear and also imparting a good finish. The broaching is done on a small hand press as only a light force is required. Fig. 19 shows a few bearings which are die castings of whitemetal without any shells of gunmetal. They are perfectly satisfactory as long as there is no pounding on the bearing, but they cannot carry heavy loads as the whitemetal will spread.

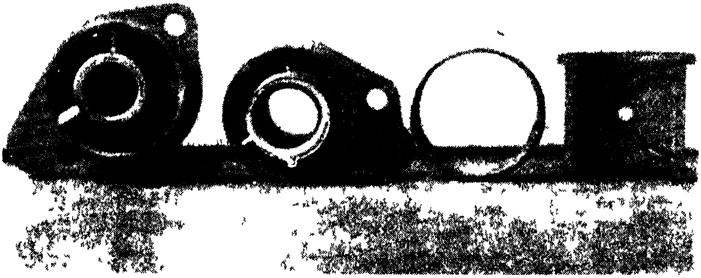


FIG 19 GROUP OF WHITEMETAL DIE-CAST BEARINGS

BEARINGS USED ON SHIPS AND TURBINES

The usual types of gunmetal bearings lined with whitemetal are used in marine engines, but the bearings for the propeller shaft to take the load and also the thrust are exceptional. In older ships the bushes in the stern tube were made of a heavy wood (lignum-vitae), as it was in contact with the water, and this was the best material then known. The water on lignum-vitae acts as a lubricant, and being a hard wood it was satisfactory. This material is now superseded by metal, and special arrangements made for lubrication. On small boats there is the standard type of greaser, but in large vessels the lubrication is carried out by pumps, and the apparatus is accessible from inside the ship. Lignum-vitae is still used on hydraulic machines, pumps, and vertical turbines. With important bearings a safety strip of bronze is put slightly below the surface, which would support the spindle in case of the whitemetal running. The bearings of turbines are often provided with spherical seatings.

MARINE THRUST BEARINGS

Owing to the enormous thrust from the propeller, special attention has to be given to the bearings to take this load. The multiple collar type was formerly standard for this kind of work. The collars were made in separate halves and inserted into the shaft, which allowed special hardened material to be used, and the bearing pieces were all made adjustable so that each took its share of the load. The standard type was of a horseshoe shape, so that any single thrust could be taken out for inspection. In another type of collar bearing, thrust was provided for forward and reverse, the top half of the bearing acted forward, and the bottom half reverse. These could be adjusted so that there was no play in the spindle. The greatest trouble was heating due to bad lubrication.

Some thirty years ago, a great advance in the design of bearings and lubricating was made by the introduction of the "Michell" bearing. A shaft in a bearing does not run concentric, as the oil film spreads and forms a taper wedge, commencing at the point of least pressure and gradually tapering off as shown in Fig. 20. The Michell bearing takes advantage of this fact, known as Reynolds' theory, and consists of pads which are pivoted, so that they can swing over and allow the oil film to form the taper wedge. The pads are of approximately square section, but they are modified to suit different conditions; for instance, the large pads in Fig. 21, each about 12 in. across, are for taking a thrust, and shaped accordingly to suit the flange. One pad is placed face downward to exhibit the spherical seat which allows the swivelling motion. There is no necessity for oil grooves of any description, in fact they would be a disadvantage owing to upsetting the oil film.

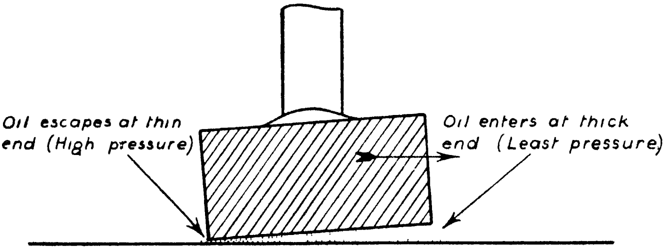

Oil escapes at thin end (High pressure)
Oil enters at thick end (Least pressure)

FIG. 20. DIAGRAM SHOWING WEDGE-SHAPED OIL FILM

The Michell bearings are designed for use either as journal or thrust bearings. To show the great advance made, a few figures will be given. With a collar type of bearing only a bearing pressure of 60 lb. per sq. in. could be satisfactorily used, but with the advent of Michell bearings, 500 lb. per sq. in. at speeds up to 100 ft. per sec. is standard, and tests have been carried out in which a pressure of 3000 lb. per sq. in. has been sustained for continued periods, and even loads of 5 tons per sq. in. have been imposed without failure.

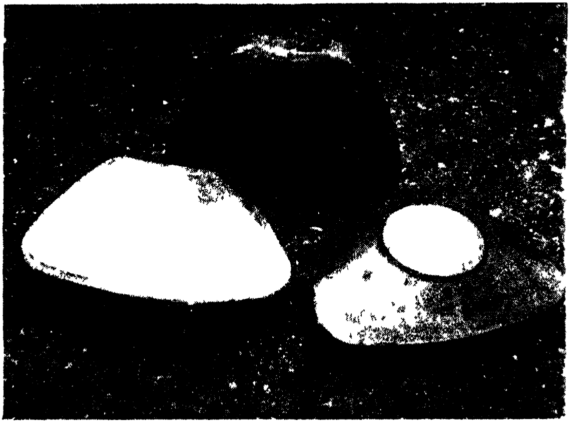


FIG. 21. MICHELL THRUST BEARING PADS

A complete assembly of a Michell bearing as applied to a turbine rotor, showing the thrust pads in position and a loose one above, is given in Fig. 22. The bearing is enclosed in an oil bath, and is attached to the frame of the turbine. The design is very compact, a further merit.

LUBRICANTS

There are hundreds of different oils for lubricating purposes on the market, but they can nearly all be classified into a few divisions, namely, mineral oils, vegetable oils, animal oils, and compounds, the terms giving the source of the products. Mineral oils are derived from such typical substances as oil shales, and are supplied from the distillation of crude petroleum. Vegetable oils are obtained from plants such as the castor oil and olive oil The animal oils are well known under such names as sperm oil from the whale, and neatsfoot oil from the feet of beasts. The compound oils are a mixture of any of those already mentioned. The bulk of oil belongs to the class known as mineral oil. There are other lubricants, such as graphite and grease, and also water, which is a lubricant for lignum-vitae.

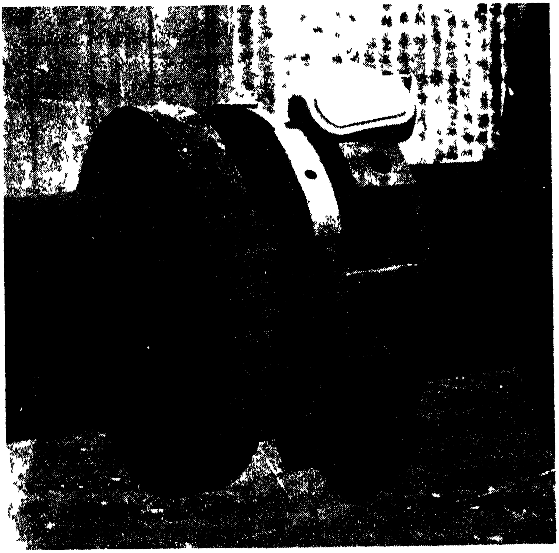


FIG. 22. MICHLLL THRUST BEARING FOR TURBINES

The lubricants have differing physical properties which are useful for special cases. One of the most important properties is viscosity, or thickness. Water or paraffin has a low viscosity, and oils, grease and treacle have higher figures. The greater the viscosity, the greater the carrying load can be, as it is harder to squeeze the thicker lubricant out, but it must also be remembered that it will lower the speed. A rough rule is that the larger the bearing the heavier should be the oil, and that as the speed increases, so should a thinner oil be used. The light oils should only be used with a light unit bearing pressure.

With any new machine, the oil should be changed after a short time, as it becomes contaminated with the sand, etc., which cannot all be removed from the castings. With machines which are affected by the weather, such as motor-cars and cycles, winter and summer conditions demand different oils. The various makers usually specify the type in their instruction manuals, and these should be carefully studied. Never mix oils, as when different oils are mixed a sludge or deposit is likely to occur and foul the complete machine, or in any case will produce hot bearings. In some compressors rubber packings are used, and as mineral oil destroys rubber, the lubricant which should be used is glycerine.

Lubricants need not be in liquid form, as besides thick grease, there are still graphite, talc, etc., which although often mixed with a liquid are sometimes used quite dry in powder form. Examples are lace-making machines and chocolate machines, and the powder is either dusted on or syringed into the bearing. Graphite in an extremely fine condition and mixed with oil is a good lubricant, and imparts a fine finish to bearings.

It has been found that if two unlubricated surfaces are moved, the amount of force required is proportional to the load acting on them, but if the surfaces are

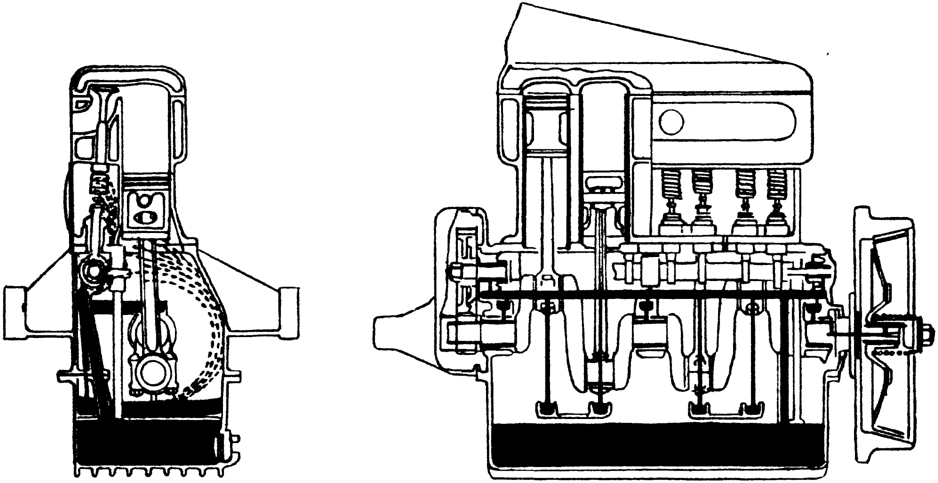


FIG. 23. SHOWING TYPICAL DISTRIBUTION OF OIL TO THE BEARINGS OF A PETROL ENGINE

correctly lubricated, the amount of force is very greatly reduced, so that much heavier loads can be carried. The action of the lubricant can be likened to that of ball bearings, the atoms of the oil sliding over each other and not allowing metallic contact. When bearings seize, the reason is because of metallic contact due to the breaking of the oil film; in fact, if a perfect film of oil could be guaranteed in the bearing, the material of the bearing would not matter. The oil film does not always work, although this is the object aimed for, and another property called oiliness comes into play. It has been noticed that with two different oils having similar properties, one was more satisfactory as it appeared to coat the surfaces where the other failed, this being a feature of castor oil.

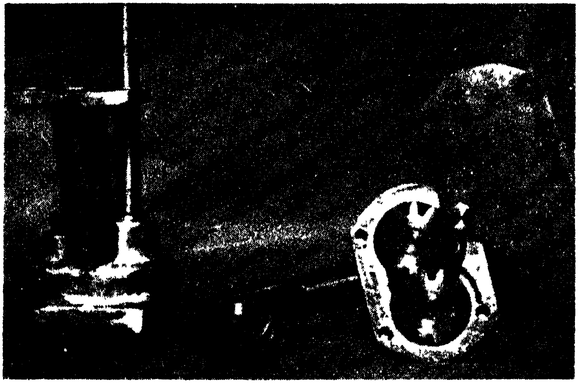


FIG. 24. GEAR TYPE OIL PUMP WITH COVER REMOVED

LOW AND HIGH PRESSURE OIL PUMPS

The lubricant is sometimes fed into the bearings by gravitation alone, but more often is pumped in under pressures varying from a pound or two to hundreds of pounds per square inch, according to each special case. The lubricating of motor-cars has been specially studied, and every detail of the engine is automatically lubricated under pressure. A few firms carry out a similar system with the complete chassis. Fig. 23 illustrates the main lubricating parts of one car engine. The oil is contained in the oil sump which has cooling fins cast on, and a pump is submerged in the oil. The lubricant is pumped up the pipe to the main distributing pipe running the whole length of the engine, and from this is fed to the main crankshaft bearings, the camshaft bearings, timing gears, clutch spigot, and also to troughs into which the connecting rods dip and fling the oil to the cylinder walls. With pressure systems an oil release valve is necessary, and it is usual to have this adjustable so that the oil pressure can be easily regulated.

The gear pump is the commonest type. Fig. 24 gives an outside view and also a view with the bottom cover removed to show the gear wheels. As the gears revolve, they suck the oil through the inlet pipe and deliver it to the opposite side, where it is forced up the outlet pipe. By fitting ball valves they can be made to operate in both directions. Backlash in the gears does not matter, in fact a small amount is an asset.

In a test carried out on a gear pump the following figures indicate the drop in pressure when the temperature is altered. At 50 degrees the pressure was 80 lb. per sq. in., but at 80 degrees the pressure dropped to 20 lb.

Two other types are the rotary pump and the plunger pump, the latter being shown in Fig. 25. In this case there are two plungers driven by eccentrics from the gear wheel. The front is the plunger proper, and the rear one acts as the valve, which thereby gives a pump with no valves, or loose balls to get out of order.

Before leaving the question of lubricating, it is well to point out the importance of oil strainers or filters. They should be fitted in all cases and should be accessible. With such parts as locomotive axles where cotton waste soaked in oil is used, this itself feeds the lubricant, and also excludes dust, dirt, and grit; but with force feed any dirt present is also forced to the bearing, unless first removed by a filter.

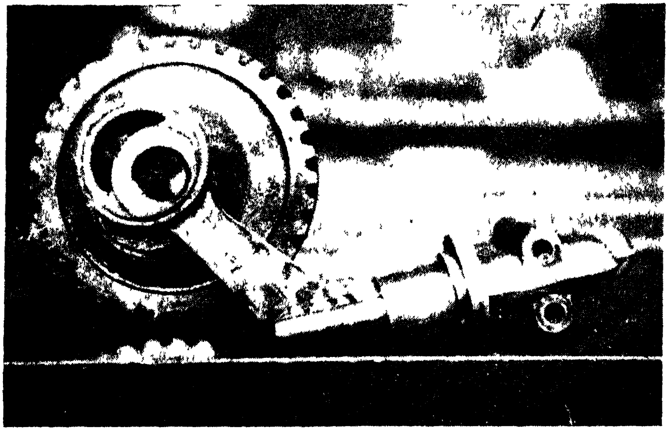


FIG. 25. PLUNGER TYPE OIL PUMP

Undoubtedly, the perfect way to lubricate is to have a supply of oil entering the bearing where the pressure is least. When this takes place the journal of the shaft will be eccentric to the bearing, the oil entering at the thick edge and leaving at the narrow edge (see Fig. 20).

OILERS (HAND AND MECHANICAL)

The simplest and best known type of hand oiler is the oilcan, and there is not much variation in the oil cups except the size. Some are only receptacles, while others have a wick feed which filters the oil and only allows a small quantity as required. In Fig. 26 a wick-feed type is illustrated. The wick is fed from the reservoir and passes through the cap, and then down over the half bearing. The amount of lubricant is under control, as the cap can be adjusted to press tightly on the wick in order to reduce the supply. Another device is the felt pad, which will hold a supply of oil, and feed it on to the bearing. A useful type of oiler for small work is the spring-well, in which the inlet is kept closed by a ball and spring until the oilcan spout is forced in.

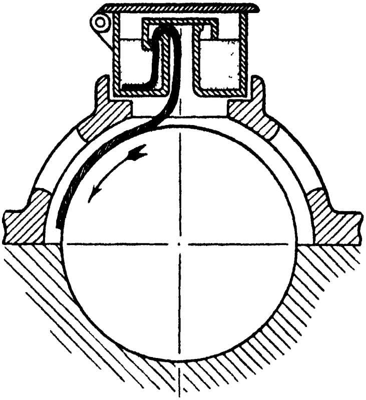


Fig. 26. Lubrication by Wick and Reservoirs

Under the class of mechanical oilers come pumps, and also such things as oiling rollers. The ways of machine tables must be lubricated, especially in such cases as planers and grinders which are in use continuously. This is done by rollers sunk in oil wells and mounted on a light spring, so that they are always in contact with the table. Whenever the table moves, it turns the rollers which carry oil to the ways and deposits a lubricating film.

GREASERS AND GREASE GUNS

With greasers we have a larger variety. There are the usual grease cups with screw-down caps, which are operated when a supply is required, and another type in which the cap carries a spring-loaded plunger, which forces the grease into the bearing as soon as there is space for it. A type of greaser used extensively on motor-cars is the nipple type, in which a grease gun is used to force the lubricant into the bearing. The nipple only has a small hole, so that water and dust cannot obtain access, and the grease gun end is coned so that no screwed connections are necessary.

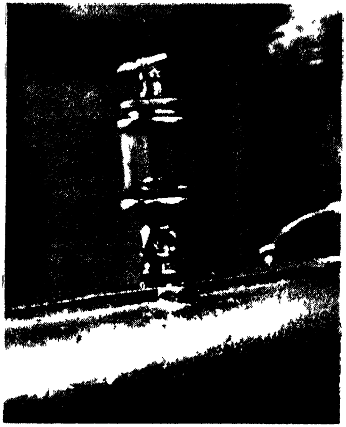


FIG. 27. DRIP FEED LUBRICATION SYSTEM

DRIP FEEDS

With those bearings where it is necessary to verify the amount of lubricant used, sight feeds are used. A standard type is illustrated in Fig. 27, and consists of a glass chamber to hold the oil and a needle valve at the bottom. The oil drips from the outlet, and can be seen and regulated by the small milled screw at the top. With the small handle at the top in the horizontal position as in figure, the needle valve is closed and no oil passes. To re-start delivery, the handle is placed vertical. Another type of drip feed is illustrated in Fig. 28, and is fitted on a compressor to feed into the cylinder. With the ordinary type the oil would be blown straight back with the pressure from the compressor, and therefore it is necessary to fit an equalizing pipe, so that the pressure in the drip feed is the same

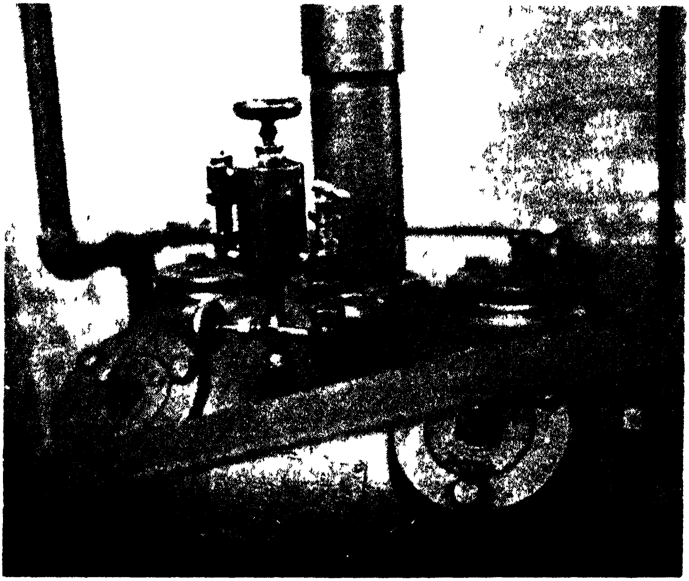


FIG. 28. DRIP FEED FITTED TO COMPRESSOR

as in the cylinder and the oil can then flow in. Fitted to many makes of motor-cycles for lubricating the engine there is a plunger pump, the discharge of which is usually operated by a spring. This pumps the oil through the drip feed, and the amount can be regulated by a milled adjusting screw, and, of course, checked by watching the amount through the glass sight feed.

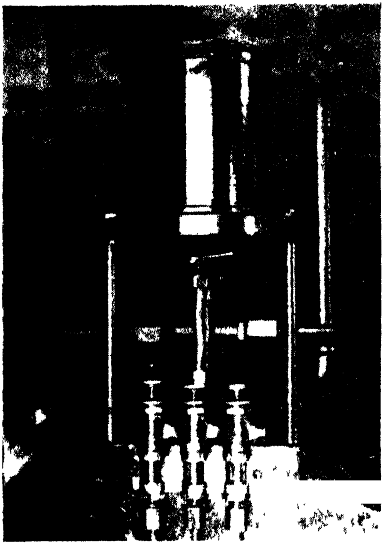


Fig. 29. Multi-drip Feed

Fig. 29 pictures multi-drip feeds all fed from a common reservoir, having three leads to different bearings on the machine. By this device, one filling and one tap controls the supply to all the bearings.

RING OILERS

Ring oilers are extensively used in machine tool work, and have already been described under bearings for shafting. The shape of the rings varies, as does also their weight, both of which are modified to suit the type of bearing. In one type of ring a channel section is used, which tends to bring up more lubricant than the plain type. The rings of oilers are usually about twice the diameter of the shaft. Another modification is the provision of scrapers and channels in the housing to take the oil from the ring and distribute to the oil groove. Instead of solid rings, chains can be used to deliver the lubricant, and Fig. 30 illustrates the use of a chain which lifts a mixture of oil and oilstone powder, and then by reason of centrifugal force delivers it to the teeth of the gears. At high speeds, chains are liable to churn up the oil and cause a foam.

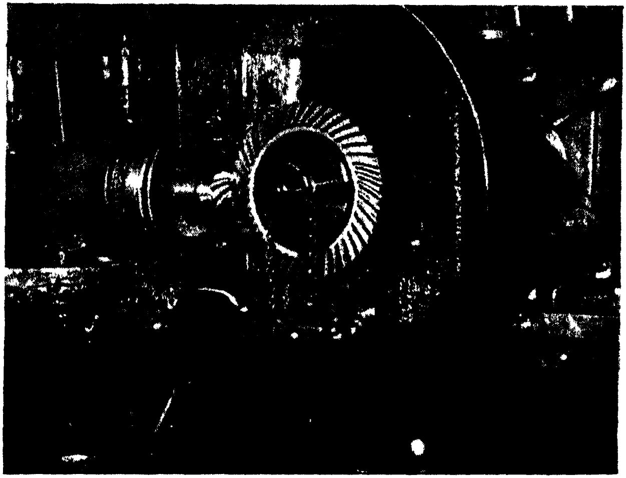


FIG. 30. LUBRICANT LIFTED BY CHAIN

OIL INDICATORS

There are many types of oil-level indicators. Some, as shown in Fig. 31, have glass tubes in which the level of the oil is seen, while others have floats which move an indicator. There are simpler forms in which an indicating rod is used, or again, cocks or taps are fixed at the various levels. Some bearings are also fitted with a thermometer to indicate the temperature, so that any special increase is seen and steps can be taken to rectify the fault.

MATERIALS

The materials of the component parts of bearings are extremely variable. The following are a few—

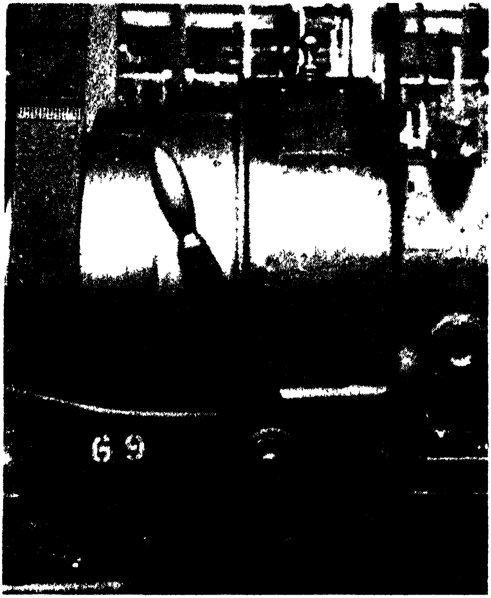


FIG. 31. LATHE HEADSTOCK FITTED WITH WINDOW TYPE OIL INDICATOR

Mild steel and whitemetal bearing.
Cast iron and cast iron bearing.
Steel and cast iron bearing.
Hard steel and phosphor bronze or gunmetal.
Hard steel and agate.
Hard steel and fibre.
Soft steel and wood.
Stainless steel and brass.
Hard steel and chilled cast iron.

Hard steel and hard steel.

Many varieties of oil-less bushes consisting of lead base, graphite, wax, etc., and also stone-lined bushes.

There would be no need for such a huge variety of materials if a film of oil was always present, as then metallic contact could not occur. The grain structure of the material is more important than the composition or analysis of the material. The best bearing metals are those having a mixture of hard and soft constituents distributed evenly throughout the material. Similar metals are more liable to seize than those of dissimilar material. Another reason for varying metals is the different lubricants, and also the bearing pressure, as this varies from 30 to 8000 lb. per sq. in. A hardened steel shaft in a hardened steel bearing will stand a pressure of 2000 lb. per sq. in., mild steel in white-metal, 500 lb. per sq. in., and cast iron in cast iron, 100 lb. per sq. in. These are only approximate, and would again vary according to lubrication, and also whether the load was constant or intermittent. The bearing area is equal to the diameter multiplied by the length, and in motor-car engine practice the bearing pressures vary from 500 to 1000 lb. pressure, according to whether splash or forced feed is used for lubricating. For bearings such as gudgeon pins, these figures can be increased to 2000 lb. per sq. in. Bearing material must therefore stand high unit pressure at high rubbing speed without wearing excessively, and the friction must be low, or otherwise a lot of power is wasted. They must also resist corrosion, and the journal should be hardier than the bearing, to facilitate replacement. Hard steel bearings are extensively used on jigs and tools, and Fig. 32 illustrates a bearing of this material where the peg of a facing cutter is located in the hardened steel bush which acts as a bearing during the cutting period.

WHITEMETAL

The most important anti-friction bearing metal is the so-called whitemetal, or Babbitt. The composition varies, but is principally tin with antimony and a small percentage of copper. Whitemetal, if examined microscopically (Fig. 33), is seen to consist of cubic crystals of a hard substance embedded in a softer matrix, and it is this arrangement which makes it so very useful. The hard crystals are evenly distributed and take the wear of the bearings, and being in a softer matrix they give, and in a very short time the surface

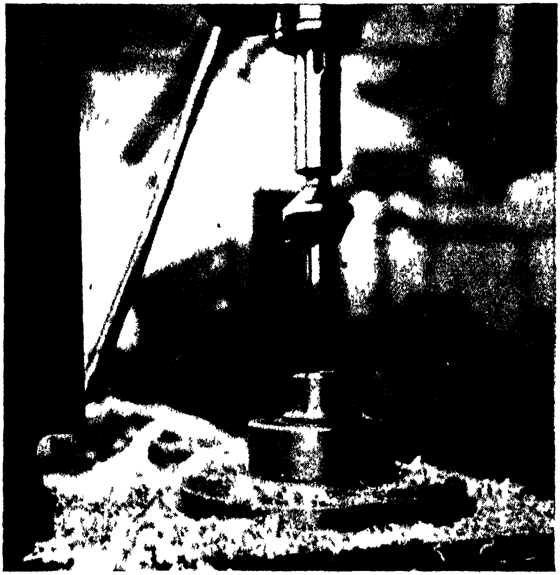


Fig. 32. Showing the Hardened Steel Bearing in which the Peg of a Facing Cutter is Located

is in a perfect condition. If subjected to impact blows, whitemetal must, however, be backed with a harder material. There is not much difference between the friction in the soft and hard metals, but with the softer metals, when run, they are soon reduced to a level and smooth surface which distributes the load over the bearing. With whitemetal there is also less chance of scoring the shafts, as if overloaded the bearing will run hot and melt. Further, if grit happens to obtain access to the bearing, it will become embedded in the whitemetal, whereas a harder bearing would probably score the shaft.

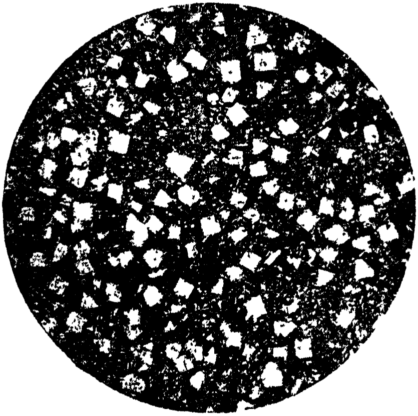


FIG. 33. THE MICROSTRUCTURE OF A TYPICAL AUTOMOBILE WHITEMETAL

BRONZE

The bronzes are much harder than whitemetals, and much more work is required to produce a satisfactory bearing. The bearings must be properly bedded to the shaft or journal, otherwise the load is taken by the high points only, which will not yield like whitemetal. The result is that heating occurs and seizure of the bearing is sure to follow, which will also probably score the journal. Fig. 34 depicts a headstock in which the bearing metal is bronze. This front bearing also takes the thrust through the steel washer, and the brasses are of square section and do not, therefore, require to be pegged.

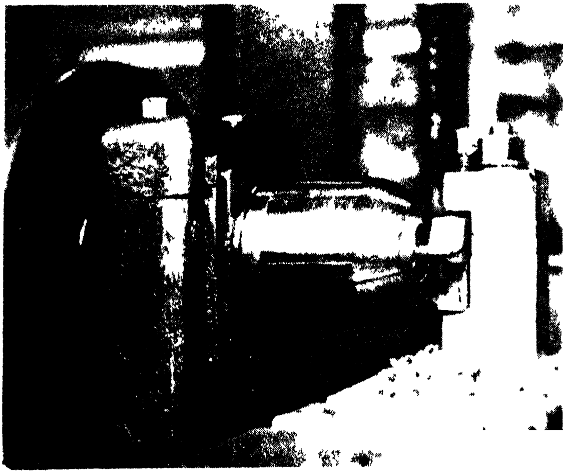


FIG. 34. HEADSTOCK WITH BRONZE BEARING

CAST-IRON AND OTHER BEARINGS

Chilled cast iron is a hard wearing metal used in conjunction with a hard steel journal, but as with the bronzes, the bearing must be perfectly scraped or bedded in. If two bearings are required to be in line, such as a lathe headstock, trouble would be experienced if both were hard bearings, and good practice is to have the front one of hard material. and the rear one of whitemetal which can be run in. The boring machine spindle in Fig. 35 illustrates the use of cast iron.

Lead acts as a good bearing metal, and lead base bearings have given good service, but they will not stand such a high compressive load as tin base bearings. The following is the composition of some railway wagon bearings: 70 per cent lead; 15 per cent antimony, and 15 per cent tin. There is also a bronze bush on the market in which graphite up to 40 per cent by volume is mixed in the metal. By reason of this the bush is porous, and will hold lubricant for some considerable time. Vulcanized rubber is the material which has been used for such parts as pumps and propeller shafts and, as with lignum-vitae, the water present in such cases acts as the lubricant.

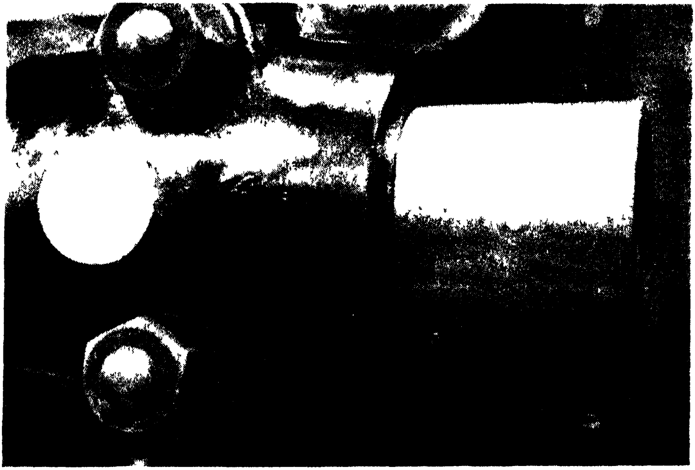


FIG. 35. BORING MACHINE SPINDLE WITH CAST IRON BEARING

OIL-LESS BUSHES

There are many different materials used in the manufacture of the so-called oil-less bushes. In some cases, wood is impregnated with wax or oil, and in others a bronze bearing is used with inserts of graphite. The lignum-vitae bushes do not require oil, as they naturally contain a resinous substance, and are usually used where water is abundant, which also acts as a lubricant. Lignum-vitae has been used mostly for the stern tube bearings of ships, and the propeller shafts were fitted with bronze liners, on which the wood acted, so that no rusting could take place. Stainless steel is now being used, in conjunction with oil-less or bronze bushes, on car parts which are not fully protected from the weather. With the oil-less bushes it often pays to add an oiler or greaser, as shown in Fig. 36. These are shackle bushes for motor-cars and besides the bush end plates are also fitted, and greaser nipples, so that a grease gun can be used occasionally. Oil-less bushes are also used for such details as loose pulleys and the rollers of conveyors, where lubrication is apt to be neglected.

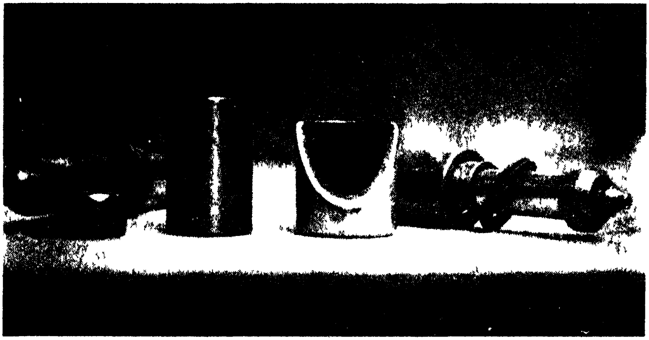


FIG. 36. SHACKLE PIN BEARINGS

WEAR OF BEARINGS

The wearing of bearings is due to breakage of oil films, and through dirt and foreign matter obtaining access. Most wear takes place at starting and low speeds, as the shaft tends to float when running at speed. If a bearing is not correctly proportioned as regards length, wear is liable to take place at the ends, as if too long the journal will act as a beam and bend. When a bearing runs hot under standard conditions, an increased diameter does not help, for though a larger surface is obtained, the velocity also increases. To obtain a cooler bearing one has to increase the length, as extra surface is then gained with no extra velocity. The part where most wear takes place is at the off-side of the bearings. This is the side of greatest pressure, and where the thin end of the oil film occurs (see Fig. 37).

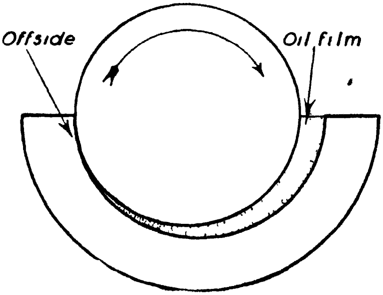


FIG. 37. EXAGGERATED VIEW OF BEARING

Explaining why the greatest wear occurs at one side

Besides machine parts and motor-cars, whitemetal bearings are now used extensively on railway work, one reason being the ease of replacement when necessary by remetalling.

The whitemetal is only a thin layer, and if the bearing runs hot and melts there is no fear of damage resulting, as the part cannot get away and break other parts of the machine. It is claimed that die-cast bearings of solid whitemetal are very good, and engines have run quite successfully on them, but there is always the risk of the metal spreading, especially at the ends, and as a thicker layer of metal is used, damage to the engine is likely if the bearing runs out. Whitemetalling is not confined to small bearings, for in one bearing from a huge crushing machine no less than 1200 lb. of Babbitt were required for relining.

CLEANING AND TINNING BEARING SHELL

The bearing must be quite clean before the tinning operation is begun, so take care that fumes during the heating process do not foul the surface, and always heat the bearing from the back. It cannot be over-emphasized that cleanliness in whitemetalling is the essential factor. Pure tin is the best solder to use, but in any case only use the very best tinman solder, and killed spirits of salts is the most popular flux. When tinned, dip the bearing in a pail of water, and wipe the surplus metal off with clean cloth. Then there is less chance of dirt, fluff, or pieces of cotton getting on the bearing face. For heating large bearings, a muffle should be used, but a gas ring or a blowpipe can be used on small bearings. This comes in handy for heating the jig, and also keeps local parts heated. If ready to proceed, the bearing should again be heated and a thin layer of best solder be given, as in Fig. 38. This tinning can easily be done by plunging the bearing into the molten metal, but there is always the risk of dirt, etc., contaminating the metal.

TESTING TEMPERATURES

If whitemetal is heated higher than necessary, the metal is useless, and it should not be repeatedly heated, and then left to cool. Only use sufficient metal to do the job required each day, and if there is any over, pour it out of the melting pot so that it is cooled quickly, and it then can be used on the next work. Two other important points are that the metal should be stirred well, so that the mixture is homogeneous, and the temperature carefully watched. A pyrometer is the correct instrument to use, but often a mechanic cannot obtain one, and must therefore be able to check this without instruments. In place of a pyrometer, small pieces of white paper can be used, which should be dipped into the metal and withdrawn. If the metal is at the correct temperature the paper will become brown, but not burnt or charred black. Another sign of over-heating is a blue ash, which is an oxide forming on the surface of the metal. Some men cover the surface of the metal with a layer of wood charcoal to prevent this oxidization. The actual temperature varies with different whitemetals, but all should be heated till they run freely, when pouring should be done.

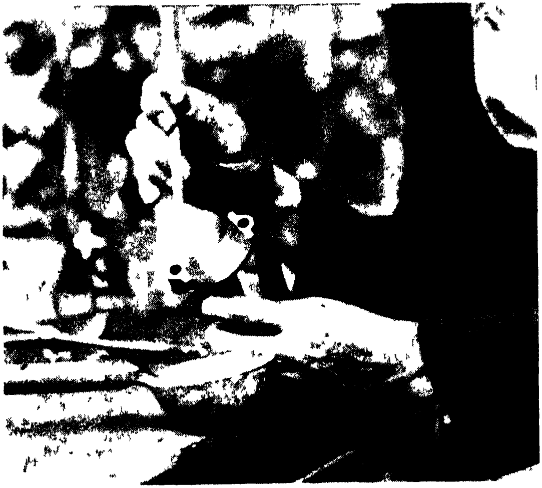


FIG. 38. TINNING A SPLIT BEARING

MOUNTING BEARINGS AND POURING

The next operation consists of mounting the bearing in a fixture and pouring the metal. We will first discuss the fixtures. If possible, make the fixture so that the metal can be poured vertically, but in all cases see that provision is made for extra supply of metal at the top. All the mandrels or pieces in contact with metal should be cleaned and smoked to prevent the metal sticking to them. Instead of smoking, the part can be dusted with French chalk. The necessary fixture can easily be adapted by the mechanic to suit each particular case. With half bearings, an angle plate can be used, and a half mandrel, or a piece of sheet metal bent to the shaft diameter instead of the solid mandrel. This is placed against the side of the angle plate, and the bearing shell clamped up in position. Where extra metal is required on the faces, either clay, putty, or millboard pulp can be used, or a special metal extension added. The clay is also useful to lute up any small inaccuracies at the various joints. Fig. 39 shows the horizontal pouring

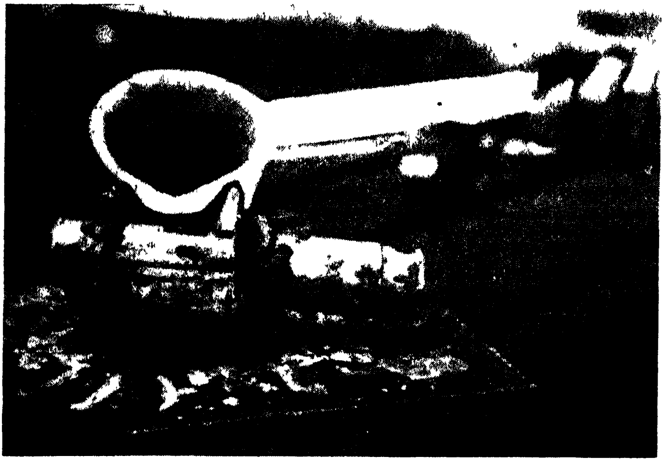


FIG. 39. POURING MOLTEN METAL OVER A METAL CORE WITH CARDBOARD END PIECES

where a shaft and cardboard end pieces are used, also plenty of clay luting to prevent the leakage of the metal.

In certain cases whitemetalling is done in position. The bearings on small engines, and lead-screw nuts on lathes, are examples. The prepared spindles are mounted in line, and the necessary packings, etc., used, and then the metal poured direct. When ready for pouring, heat the bearing and the fixture until the tinned surface begins to run, and then hold the ladle as near as possible to the bearing and pour in a steady stream, without any splashing and in one go. Always allow plenty of metal, as surplus can easily be turned off, while cavities cannot be filled. The bearing should be cooled from the bottom, while the top is kept hot so that the metal can flow down the bearing to fill up any spaces or gas cavities. To help this, dip a clean piece of wire in and out of the part still molten, which will tend to burst the cavities and allow the still molten metal to flow in. This period is only for a minute or two.

In workshops where production is large, whitemetal die-casting machines are used, and the metal is forced into the die by means of a plunger. Whitemetal bearings have also been successfully manufactured by centrifugal casting.

TESTING AND DEFECTS

A good bearing that has contact with its shell will ring true when lightly struck with a hammer. When scraping the metal, if the material seems hard and gritty it is a sign of overheating, and the bearing should be scrapped.

If a bearing does not ring true, or the metal has not adhered to its shell, it must be relined, and greater care taken to see that the flame does not reach the tinned surface, and also that the shell is heated enough to run the tinned surface before pouring the metal. Blowholes are often due to the temperature of the metal not being correct, or to bad pouring; but they can also be due to oil, water, or air being in the bearing, and unable to get away.

OILWAYS AND CLEARANCE

Oilways carry the lubricant from its source of supply to the bearings, and consist of either pipes, channels, or drilled holes, according to the design of the machine. Oilways also collect used oil, and return it to be filtered ready for circulation. The area of the pipe should be sufficient to flood the bearing with oil, and the mechanic must see that all oilways are clear. Many ingenious devices have been made to supply oil to revolving cranks of steam engines, and one which is well used consists of a sight feed held vertical by balance weights.

The line drawing illustrates standard practice in auto engine design with regard to oilways (Fig. 40). The oil from the pump is delivered to the main distributing pipe, and then fed to the crankshaft bearings. The crank is drilled, and the oil fed via drilled holes from the bearing to the throws, which lubricates the connecting-rod big end. The various bearings, oil pipe, and oil holes can be seen in the photograph, Fig. 41, which shows the case with the components removed. So that no leakage of oil will occur at the face when the crank protrudes through the case, an oil thrower is turned on the shaft, which slings the oil back into the case. Even if any escapes past this and tends to ooze out, an oil return groove will again return it to the oil sump. In other engines the oil is fed to the crank pins by pipes, or again by means of channels cast in the crankcase. Bearing clearance is the amount of space which must be left not only to allow for an

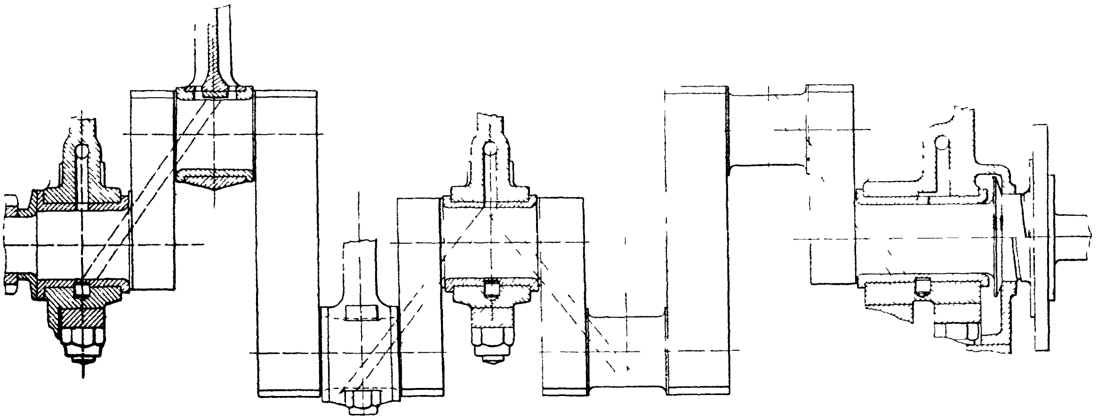


FIG. 40. THE BEARING ARRANGEMENT ON A TYPICAL FOUR-THROW CRANKSHAFT

oil film, but also for any expansion that takes place during the running of the engine and machine.

No definite figure can be given to suit all cases, and this is often left to the fitter or erector to decide. An

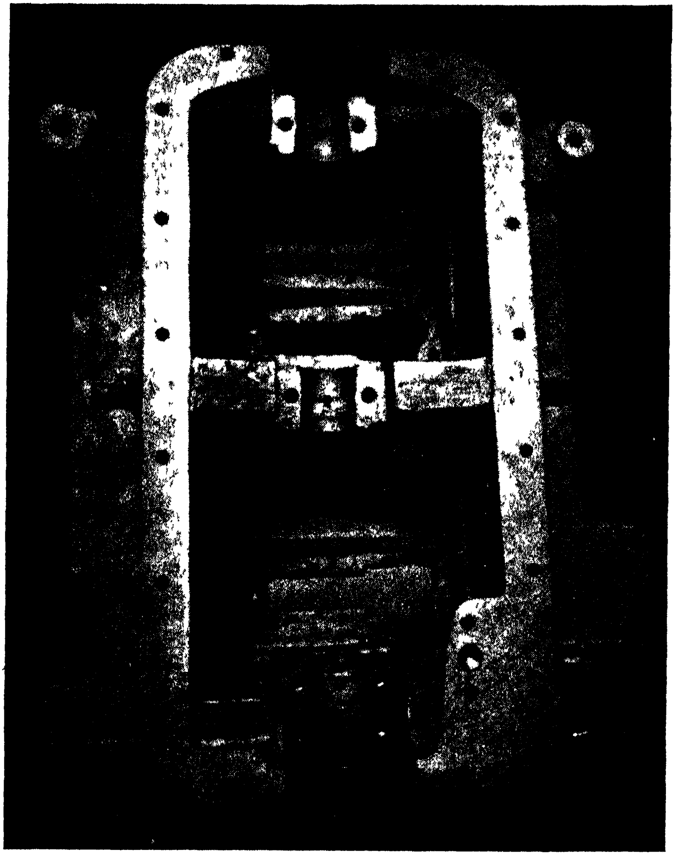


FIG. 41. LOWER HALF OF CRANKCASE STRIPPED OF THE CRANKSHAFT ASSEMBLY

average clearance for running fits on medium size shafts is to allow 0·001 for each inch in diameter. The clearance and limits for various sizes of bearings, so that a good film of oil can be assured, are as follows—

For the Shafts—

Up to 1 in. diameter the size should be minus 1 to minus 2 thousandths
,, 2 ,, ,, ,, ,, ,, ,, 1¼ ,, 2½ ,,
,, 3 ,, ,, ,, ,, ,, ,, 1½ ,, 3 ,,
,, 4 ,, ,, ,, ,, ,, ,, 2 ,, 3½ ,,
,, 6 ,, ,, ,, ,, ,, ,, 2¾ ,, 4¾ ,,
,, 10 ,, ,, ,, ,, ,, ,, 3¼ ,, 5¾ ,,

For the Bearings—

Up to 1 in. diameter the hole should be plus ½ minus ¼ thousandths
,, 2 ,, ,, ,, ,, ,, ,, ¾ ,, ¼ ,,
,, 3 ,, ,, ,, ,, ,, ,, 1 ,, ½ ,,
,, 4 ,, ,, ,, ,, ,, ,, 1¼ ,, ½ ,,
,, 6 ,, ,, ,, ,, ,, ,, 1½ ,, ½ ,,
,, 10 ,, ,, ,, ,, ,, ,, 1¾ ,, 1 ,,

OIL GROOVES

The oil grooves distribute the oil over the various faces on the bearings, and a lot of attention is now being paid to these points. The old method was to cut a spiral groove straight round the bearing, without any theory except to distribute the lubricant. The idea of an unbroken film, and also the introduction of Michell bearings with their taper wedge of oil, has altered these ideas, and now the oil is led in at least pressure, via an oil groove at right angles to the moving part. Oil grooves should have round edges, should be as few as possible, and must not be allowed to run out of the bearing, as otherwise the oil will take the easiest path, and not do its work. Oil grooves can be put either on the shaft or in the bushes, but the usual way is in the bushes, as these are the softest material. The grooves, if spiral, are cut right or left hand, or sometimes both, according to the position of the oil inlet (see Fig. 42). The faces of thrust bearings are lubricated by eccentric oil grooves which distribute the lubricant over the face. Oil grooves are also necessary on such faces as the ways or slides of machines, the underside of a grinding machine table being shown in Fig. 43. Oil grooves can be overdone, and also cut the wrong way, with the result that oil is returned and not supplied to the bearing. With whitemetal bearings,

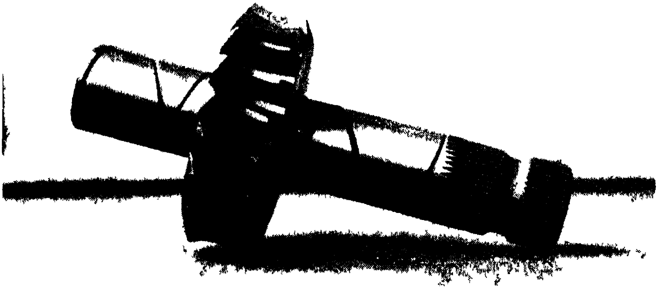


FIG. 42. OIL GROOVE ON WORM WHEEL SHAFT

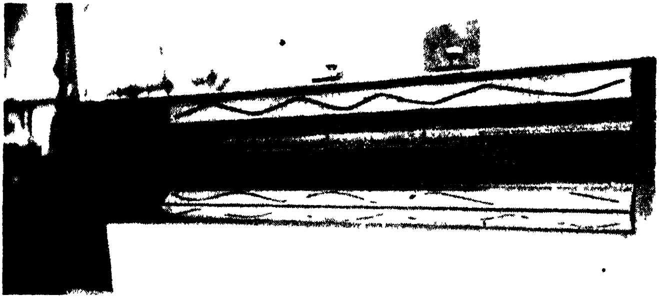


FIG. 43. LONGITUDINAL OIL GROOVES ON UNDERSIDE OF GRINDING MARKING TABLE

grooves known as mud grooves. consisting of a chamfer along the butt faces of each half bearing, are added. These, of course, should be of a slow chamfer, so that the oil can be drawn in by a wedge action, and the grooves must not extend the full width of the bearing. Oil grooves in the region of high pressure are not only bad, but often harmful, as they break up the film of oil. The oil grooves can be cut on a lathe with a screw-cutting attachment, or by means of oil grooving machines. If no machines are available, the fitter can easily cut the grooves by hand.

On motor engines it is often found necessary to fit baffles at the entrance or bottom of the cylinder bores to reduce the amount of oil splashed upon them.

If more oil than is necessary for lubrication is delivered, a certain amount will get burnt, or charred, and form carbon deposit; which results in pre-ignition, failure of piston rings, foul ignition plugs, and many other troubles due to contamination of the oil.

FINISHING SURFACES

The surfaces of bearings should be in a perfect condition if friction is to be reduced. With steel shafts the journals can be ground to size, but although the surface appears smooth, if examined under the microscope, it appears more like a ploughed field. One of the places on motor-car engines where the finish of the bearing surface has improved is the cylinder bore, and the majority of bores are now rolled, lapped, or lined. A rolling tool is shown in Fig. 44, and consists of hardened rollers free to revolve on a spindle which is fixed into the bore, and while rotating is reciprocated up and down the bore. It not only produces a good finish, but slightly compresses the metal and gives it a harder surface. Honing is another method. The hone seen in Fig. 45 consists of fine stones mounted in a holder, and the action is similar to grinding except that only small amounts of material are removed. After the surfaces have been attended to, there is the scraping and bedding-in of the bearings, working clearances, and also the alignment of the bearings A frequent cause of failure in bearings can be attributed to lack of alignment, or a bent shaft which unduly stresses the bearing. If a shaft is weak, or the distance between the supports too great, similar trouble will be experienced. All these points should therefore be watched during fitting and assembly. With connecting rods for petrol engines, this is always checked by the assembler. Fixtures are provided to test the alignment and, if necessary, to reset them correctly.

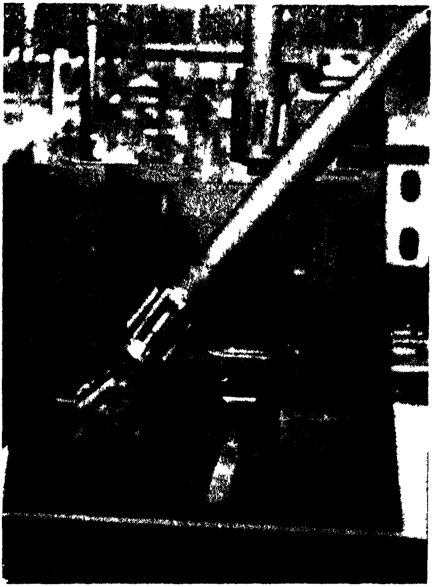


FIG. 44. ROLLING TOOL

The fitters can be saved a lot of work during the assembly if all bearings are reamed in line and broached as depicted in Fig. 46. The type of broach with rounded burnishing teeth can also be used in many other parts for finishing round holes, such as valve guides, tappet guides, etc. Often the working clearance is given by means of packing shims, and this is the system used in the majority of connecting rods. The bores of

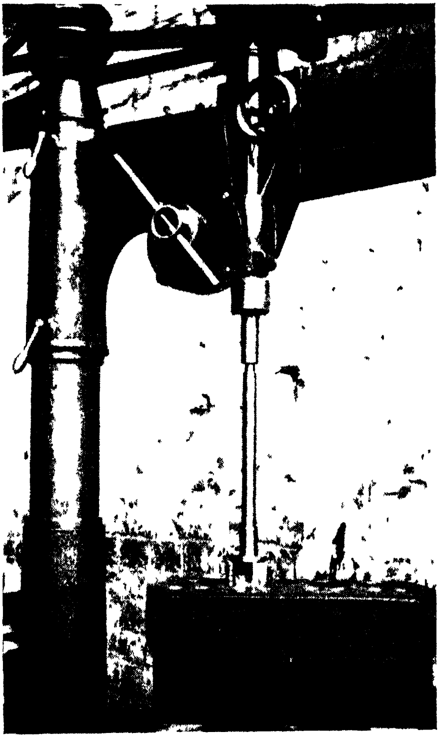


FIG 45. HONING TOOL

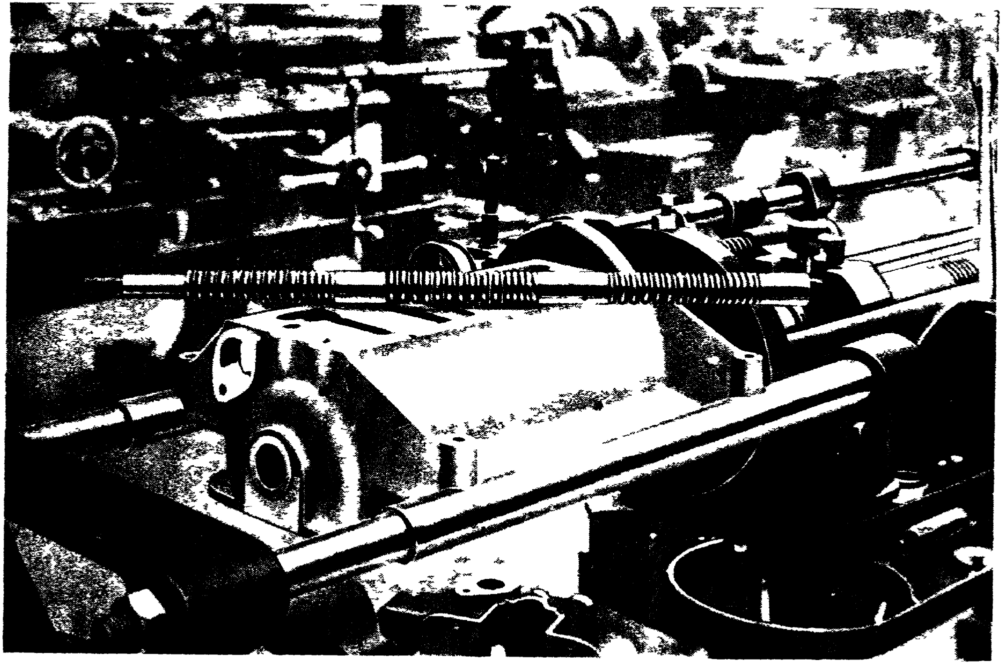


FIG 46. BROACHING A THREE BEARING CRANKCASE

whitemetalled connecting rods, and in a few cases even the main bearings of crankcases, are broached to size before bedding-in. When dealing with whitemetals, the last teeth of the broacher are rounded and act as burnishers or rollers, compressing and hardening the surface.

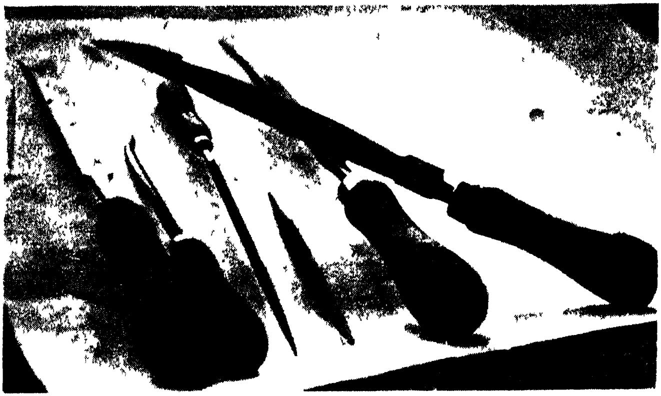


FIG. 47. GROUP OF BEARING SCRAPERS

SCRAPING AND BEDDING-IN BEARINGS

When the shaft is true, correct to diameter, and has a good finish, the operation of bedding-in can be commenced. An an illustration, we will take a three-bearing crankshaft for a petrol engine. First, the tools must be described. They consist of hardened steel with cutting edges, and their shape depends upon the articles being scraped. Flat ones similar to chisels are used for flat surfaces, and round ones for journal bearings, as seen in Fig. 47, which illustrates a collection. They are easily made out of old files, and the edges should be kept sharp. Even when in use, an oilstone should be kept

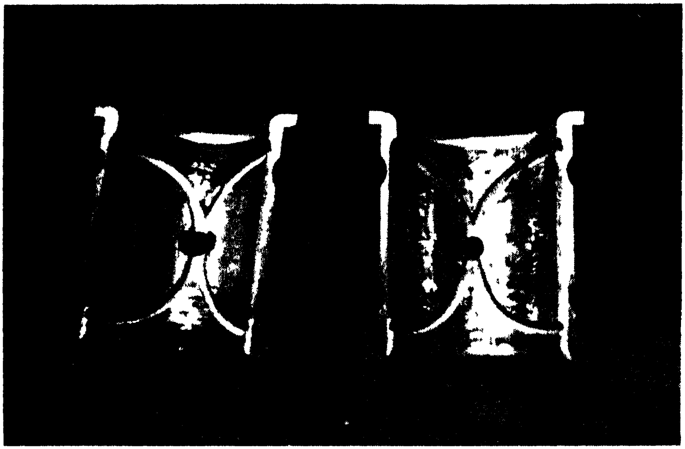


FIG. 48. BEARING JOURNALS BEFORE (LEFT) AND AFTER (RIGHT) SCRAPING

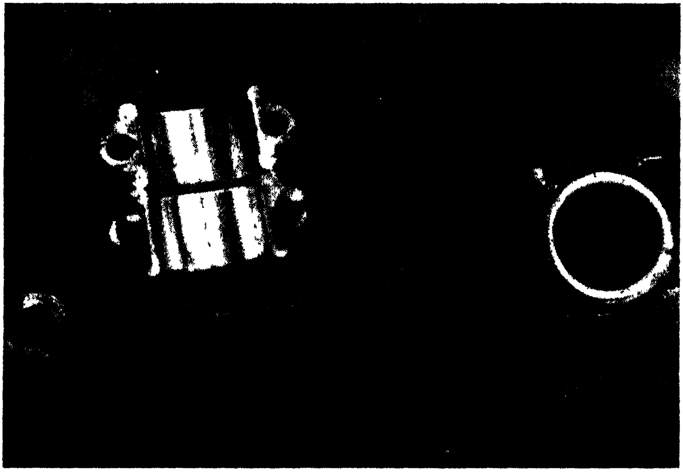


FIG. 49. TOP OF CONNECTING ROD BEARINGS

handy so that a few light rubs can be given. The half bearings in the case are first put in position, the cap bearing in all cases being left till later. The crank or standard bar is then lightly smeared with a marking paste, such as Prussian blue or red lead, and then laid in the bearing and revolved while being held by pressure in the bearings. When lifted out of the bearings, they are marked on the high places. The various scrapers are now taken, and the high points reduced by removing very thin shavings of metal. The process of scraping is more difficult than the description sounds, and experience is necessary before good work is produced. The operation is illustrated in Fig. 48, where two steps are shown, the left hand one being marked ready for scraping, and the right hand one seen after scraping. The marking of the bearings and scraping is continued until the marks are distributed over the whole bearings.

When there are three bearings, much more care is needed, and the centre bearing should not be placed in position till the two end ones have been finished. The next procedure is to fit the caps, and here two operations are necessary: first, the bearing surface needs scraping, but at the same time the faces at the junction of the two halves need levelling and also reducing, so that the cap is in contact with the shaft. The tightness of the shaft should not depend upon the tightness of the nuts, but on the clearance allowed when fitting. With whitemetal bearings the shafts should not be free enough to be spun round, but with phosphor bronze bearings, which will not give like whitemetal, the shaft should have no play, but be free enough to spin round. Fig. 49 illustrates a connecting rod and bearing in which the fit depends not on packing shims, but the fitting of the two halves until the correct clearance is obtained.

RUNNING-IN BEARINGS

Whitemetal bearings should be run-in for a short period. and then the bearings can be checked again. This running-in allows the whitemetal to form a perfect bearing, the small cubic crystals of hard material sinking into the softer mass, and although the bearings were slightly stiff before, they should be free after running-in. This, of course, must not be done with bronze bearings as the material is too hard, and the bearings must be free to start with.

When running-in see that a plentiful supply of lubricant is available, and control the heat of the bearings also, which can easily be managed by a stream of oil flowing over them. In replacing bearings, watch the oil holes, and see that they register with the supply, and, if remetalled, see that no metal has got in the pipe or choked the supply in any way. After fitting bearings, see that they are protected from all dirt and dust which causes the wearing of the surfaces, and, lastly, see that a bearing is kept properly lubricated with correct lubricant either continuously or at regular intervals.

SECTION XXX

BALL AND ROLLER BEARINGS

BY

A. W. JUDGE, WH.SC., A.R.C.S., A.M.I.A.E.

SECTION XXX

BALL AND ROLLER BEARINGS

INTRODUCTORY

ALTHOUGH the plain type of journal and thrust bearing has been developed to a fair standard of efficiency by careful attention to such factors as the choice of suitable bearing metals, the proper bearing load pressures and the correct means of lubrication, the fact remains that the plain bearing depends upon sliding resistance of two solids separated by an oil film. As tests have shown conclusively, the efficiency of such bearings has reached its limiting value in the case of whitemetal bearings and hard steel shafts, or journals, running in oil fed under high pressure through suitable oilways. In the search for still further improvement in bearings, more particularly in the case of motor-car and cycle ones, it was discovered that by substituting rolling friction for sliding friction in bearings a considerable reduction in friction could be obtained. To appreciate the principle underlying the action of a ball or roller bearing, one may consider a heavy metal object having a flat lower surface resting on a flat metal table—the " marking-off " table of the engineer's shop, for example. If one attempts to drag this object across the table a certain appreciable effort will be required. This effort will be less if the table top is greased or oiled. If, however, one places metal rollers under the metal object, as shown in Fig. 1, the force required to move it across the table will be found to be appreciably less. The approximate forces required to move the object under the above conditions are indicated in the diagrams (Fig. 1). By lubricating the rollers and the top of the table a still smaller effort will be found to be required.

BALLS AND ROLLERS

If the rollers are replaced by hardened steel balls a somewhat similar saving in effort is obtained, but if the object to be moved is relatively very heavy, unless a large number of balls is provided, the balls being in contact only at points may indent the surfaces or themselves be distorted.

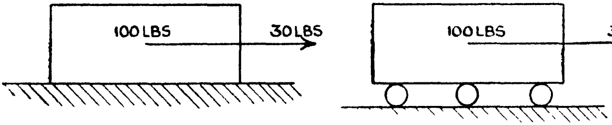


FIG. 1. ILLUSTRATING (LEFT) SLIDING, AND (RIGHT) ROLLING FRICTION

We thus see that for heavy loads rollers are better, whilst for light loads balls may be used satisfactorily. In the same way one always uses either double row ball bearings or roller bearings for heavy journal and thrust bearings in engineering practice.

THE ADVANTAGES OF BALL BEARINGS

The engineer having to handle, fit, adjust, and maintain in service ball and roller bearings should certainly be acquainted with the more important properties and advantages of these bearings. We have already referred to the reduced friction of ball and roller bearings compared with plain ones, and in this respect cannot do better than quote from some interesting tests made on Hoffmann ball bearings as used for line shafting hangers.

A shaft $2\frac{1}{2}$ in. diameter was supported at each end in plain gunmetal bearings, one set being fitted with needle lubricators, and a second shaft with two oil rings for automatic lubrication; another $2\frac{1}{2}$ in. shaft had Hoffmann ball bearings. A total load of 2,300 lb. was supported in each case. The tests were carried out for several hours so as to obtain steady conditions. Measurements were made of the friction in each case, and it was found that at 250 rev. per min. the two sets of plain bearings gave frictional values which we will represent by the numbers 160 and 130 respectively, whereas the ball bearings gave only 15.* These results prove definitely that the friction of the ball bearing was only about one-tenth that of the plain bearings. A similar statement applies also to the low frictional losses experienced in roller bearings.

The amount of power wasted in line shaft transmission where plain bearings are used is considerable. Although some authorities put this loss as high as 50 per cent of the power transmitted, we are inclined to be a little more conservative and accept the loss figure as 25 per cent, i.e. one-quarter of the power transmitted in line shaft bearings. By substituting ball bearings for the plain ones, this loss of power, as we have shown by the test results quoted, is reduced to one-tenth of its value. It will thus be seen that only one-fortieth of the power transmitted, instead of one-quarter, is lost by friction at the bearings; this represents a big saving in power.

Quite apart from their greatly reduced friction, and therefore saving in power, ball bearings are much *more compact*, occupying only a small width, although of greater overall diameter than plain bearings.

They are *more reliable*, for if protected against dust and dirt, and properly lubricated once in a while, they are immune from breakdown. *They are cleaner*, for the lubricant consists of a special grease that does not run out, and, unlike plain bearings, there is no constant flow and drip of oil over adjacent parts. In industrial factories—more particularly in those engaged in the preparation of foodstuffs, chemicals, and in textile work, this enhanced cleanliness is a big advantage.

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* The coefficients of friction in the three cases were, respectively, ·016, ·013, and ·0015.

As regards *lubrication,* there is a considerable saving both of lubricant and of time. It is only necessary, in the case of line shafting and many machines fitted with ball bearings, to pack the bearings with grease when they are first fitted, and then to inject grease—using a grease gun—once every six months. The saving effected in this respect over the cost of daily lubrication work and oil for plain bearings is about 75 per cent, according to figures furnished by Skefko bearing users.

Another advantage with ball bearings is that *much higher speeds* are feasible in many cases; for example, with line shafting, so that maximum production and economy can be effected; there is also a greatly reduced risk of bearings running hot and seizing up.

BALL BEARING LOADS AND SIZES

In what follows we shall not attempt to deal with the theory of ball and roller bearings, or to discuss the results of a large number of investigations that have been made by various authorities, but shall give just sufficient information on the subject to enable the reader to ascertain the best type of bearing to use for each particular job, and to find out the most suitable size, method of fitting, adjustment, and maintenance.

The earliest form of ball bearing was that illustrated in Fig. 2 *A*, and consisted of a pair of annular cylinders made of hardened steel with a row of balls between; as many balls as possible were used. This "two-point" bearing is the simplest type of a rolling friction bearing, but it has the drawback that there is nothing to prevent the balls from moving endwise ; this bearing, therefore, will take no end load.

In order to overcome this drawback, a " three-point " bearing of the type shown in Fig. 2 *B* and *C*, was next devised, so that the two points of contact of the ball on the grooved member were on the same radius from the shaft centre. With this type there was a tendency for the grooves to flatten at the places of contact, so that instead of there being point contact—as theory requires—there was surface contact, and consequently a sliding action. After various other possible forms of ball bearing, including " four-point " contact and grooved races (having grooves equal in curvature to the curvature of the balls themselves), the final form of journal ball bearing adopted was that illustrated in Fig. 3. In this case the ball races are grooved to a larger radius than that of the balls themselves. Prof. Stribeck, one of the earliest authorities on ball bearings, who made a large number of experiments on behalf of the German

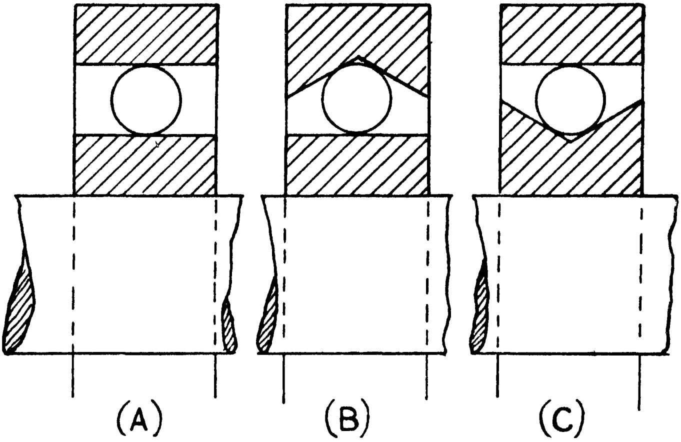

FIG. 2. EARLY FORMS OF BALL BEARINGS

Small Arms Factory, recommended that the radius of each of the grooves of the ball races should be from $1\frac{1}{8}$ to $1\frac{3}{8}$ times the radius of the balls.

There is another important matter in regard to the rubbing of the balls themselves, for if they are packed into the space between the races so as just to touch, then when the inner race, say, is rotating, the balls will roll around the outer race, and as each ball will be moving in the opposite direction to its neighbour at the point of contact with the latter, there will be undue sliding or rubbing between the balls, causing them to wear more quickly. In order to overcome this difficulty the balls are separated from each other by means of a separate member termed the "cage." This is usually a skeleton structure of soft steel or bronze, having apertures or spaces for the balls (Fig. 4). Other examples of cages will be found in the illustrations of different commercial makes of ball, roller, and thrust bearing reproduced hereafter.

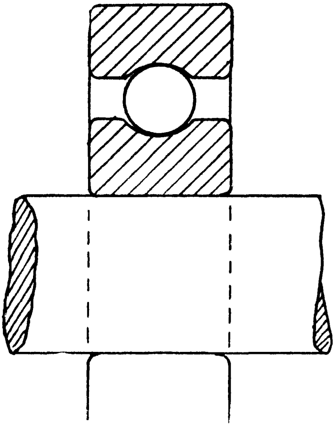

FIG. 3. THE MODERN BALL BEARING

In regard to the question of suitable sizes of ball bearing to carry definite loads, although one can calculate the size of ball to suit a given load, there is the question of speed of rotation and fatigue of the material. It is always advisable, therefore, to accept the manufacturer's recommendations for the correct sizes and types of bearing to suit specified conditions. Each ball bearing manufacturer issues, in tabular form, very complete information of ball bearing sizes (together with the actual dimensions of the ball bearings, for the designer's use) and the safe working loads for these.

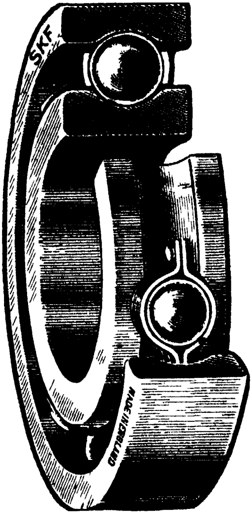

Fig. 4. The Skefko Deep Groove Ball Bearing, Showing the Ball Cage

In this respect it may be added that the safe load depends primarily upon the diameter of the ball used in the bearing, and that the load carried increases with the square of the diameter. Thus, according to Prof. Stribeck's values, a ⅛ in. hard steel ball has a safe load of 43 lb., a ¼ in. ball 175 lb., a ½ in. one of 700 lb., a ¾ in. one 1585 lb., and a 1 in. ball 2830 lb.

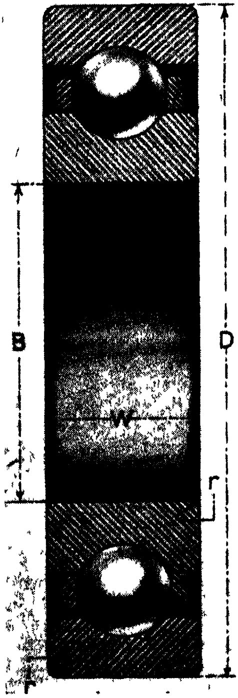

FIG. 5. THE HOFFMANN SINGLE ROW JOURNAL BEARING

TYPES OF JOURNAL BALL BEARING

There are a number of different types of journal ball bearing used in engineering work, each type having its own particular application. The simplest of the commercial types is the single row journal ball bearing, as shown in Fig. 5. These are manufactured in a wide range of sizes and in three distinct types, known as the light, medium, and heavy types. In the case of the Hoffmann standard bearings, the sizes in the light type range from those suitable for fitting over ½ in. diameter shafts up to 10 in. diameter shafts, the corresponding safe working loads ranging, respectively, from 180 b. to 2320 lb. at 1000 rev. per min. for the 3¼ in. diameter size. At lower speeds rather greater, and at higher speeds rather smaller, loads are carried.

In the medium type, the shaft sizes range from ¼ in. diameter up to 6 in. diameter, the corresponding loads at 1000 rev. per min. being 260 lb. up to 2250 lb. for the

2 in. diameter size. The larger sizes of bearing carry proportionately greater loads, but the exact values depend upon the nature of the work and method of mounting.

In the heavy type, the shaft sizes range from 17 mm. (0·67 in.) up to 100 mm. (3·94 in.), the safe loads varying from 720 lb. at 1000 rev. per min. in the former case up to 2090 lb. for the 40 mm. (1·58 in.) size. The same remarks apply to the larger sizes of bearing as to the medium type.

Other types of journal ball bearing of the single row type available are the extra light type ($1\frac{1}{2}$ in. to 12 in. shaft diameter), the magneto type (5 mm. to 19 mm. shaft diameter in fifteen different sizes), small journal bearings ($\frac{1}{4}$ in. to $1\frac{1}{4}$ in. shaft diameter in nine different sizes) and motorcycle ball bearings.

The Skefko single row, deep groove bearings (Fig. 6) are made in the narrow, medium, heavy, and standard types, and cover a similar wide range of sizes to those previously given. This type of bearing is characterized by deep uninterrupted raceways, which enable it to carry thrust loads in either direction in addition to radial loads. There is no ball filling slot in this type of bearing, so that either face can be arranged to face the load, the bearing thus being fool-proof in assembly, and reversible in operation.

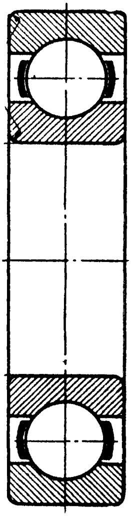

FIG. 6
THE SKEFKO DEEP GROOVE SINGLE ROW BALL BEARING

THE MOUNTING OF BALL BEARINGS

From all points of view it is essential that ball bearings be fitted to the parts in, or on, which they have to run, in the correct manner. In the past the instances of ball bearing failure investigated have almost always shown that the cause of the trouble has been due to bad fitting and incorrect mounting. Since the successful running of a ball bearing depends upon the manner in which it is mounted, we shall give rather more attention to this subject.

ACCURATE MACHINING ESSENTIAL

In the first place, too much emphasis cannot be given to the matter of accurate machining of the parts for taking the ball bearings, namely, the shaft and the housing. The latter items should be turned or ground perfectly cylindrical, with their shoulders, flanges, or abutments quite smooth and square with the shaft. All recesses, flanges, and collars should be concentric with the axis of the shaft. Split housings, collars, or adapters should be avoided, for these, if badly fitted, are apt to cause distortion of the races.

It is possible, although not always desirable, to employ split housings or bushes when fitting ball bearings under difficult conditions. Where such conditions demand a split housing the cap should be registered or dowelled, and the recess for the bearing machined with the cap bolted in position.

ALUMINIUM HOUSINGS

If the bearings are to be fitted in an *aluminium housing* it is desirable, where heavy loads are concerned, that the outer race should be housed in a liner of some harder metal. The tendency with aluminium housings is for the relatively soft aluminium to be hammered or compressed, thus causing the hard outer race to creep and finally to rotate in the aluminium housing ; the play thus developed may rapidly damage the rotating parts, more particularly as the alignment is altered. If the shaft is connected to a gear wheel, the teeth of the latter will then not mesh properly with those of the driven gear. We have experienced instances of this particular trouble in the case of the bevel-drive shaft bearing housing in the aluminium casing of a light car back axle. Fracture of the teeth of the bevel pinion and crown wheel occurred, and on examination it was found that the outer races of the bevel pinion had actually been rotating in their housings in the aluminium casing.

THE CREEP OF BEARINGS

The creep of the "fixed" member of a ball bearing that sometimes occurs is not due to the friction of the bearing as a whole, but to bad fitting, machining, or unsatisfactory material of the housing.

The conditions to be fulfilled by a journal bearing are such that *the load can never be evenly distributed around the race.* In normal examples of journal bearings the load is vertically downwards, and is taken on one portion (of about 120°) of the outer fixed race, whilst on the inner revolving race this point of maximum load will constantly be changing as the shaft rotates. It is therefore the rotating race of a journal bearing that tends to creep. When the outer race material is not soft there is no tendency for it to creep; indeed, in many cases the outer race is merely a push fit in its housing and free to move sideways by a limited amount.

VIBRATION AND BALL BEARINGS

Where creeping becomes pronounced in a stationary journal race it is an indication that the revolving mass is out of balance. This means that the point of greatest load travels round the stationary race, pressing it against the bore of the housing and causing creep. The effect on the bearings of a machine resulting from want of balance is not always fully appreciated. To secure steady running with a minimum of vibration, high-speed machines should be properly balanced, both when stationary and when running at normal speeds, i.e. both statically and dynamically.

The best method of preventing creep is to make the revolving race of a journal bearing an " interference " fit (or tight press fit) on its seat. Given reasonable care in the machining of the shafts and fitting of the bearings, there should be no difficulty whatever in this, and creep can be prevented with the heaviest loads. Keys and similar devices are quite unsatisfactory in this respect, and quickly wear away under the constant chafing.

PROPER FIT ALLOWANCES FOR BALL BEARINGS

Before discussing the subject of the practical mounting of ball and roller journal bearings it will be necessary to consider the types of fit, i.e. the machining allowances that must be employed to give the best working conditions. The usual type of ball bearing consists of an inner and outer hardened steel race, the balls and their cage. In all journal type bearings the fixed or *stationary ball race is made a sliding fit* on its shaft or in its housing. On the other hand, the *rotating member of the bearing must be made a tight press fit* (*or interference fit*) on its shaft or in its housing, as the case may be, in order to prevent creep. Each ball bearing manufacturer issues instructions or tables giving the allowances of the shafts, bearings, and housings for their particular types. The following information is applicable to most types of journal bearing.

In the case of the interference fits of *light type* journal bearings up to 4 in., and *medium* and *heavy type* bearings up to 2 in. shaft diameter, should the race be on the high limit and the shaft on the low limit the dimensions will be exactly the same in both cases, so that the fit will be too easy. On the other hand, when the bearing is on the low limit and the shaft on the high limit there will be a difference of 0·001 in., which will give too great a fit allowance for these bearings. In actual practice, however, it will be found that these extreme cases rarely occur. The majority of bearings are on the low side of standard in the bore and, as the shafts are likely to be well within the limits, in most cases the fit will be correct. In other cases, another bearing can be selected, or if the shaft is on the high limit it can be reduced slightly.

TABULAR INFORMATION ON BEARING FITS

The following tables will be found very useful in connection with machining shafts and housings. If the dimensions or allowances given are adopted by those who have to machine parts for taking ball bearings, the most satisfactory results will be obtained.

In connection with these tables it will be observed that Table I refers to the *inner races* of *both* ball and roller journal bearings (Hoffmann), and Table II to the outer races.

TABLE I

INNER RACES OF BALL AND ROLLER BEARINGS

Type of Bearing	Bore of Inner Race (Inch and Metric Sizes)	Diameter of Shaft	
		Interference Fit (Rotating Shaft)	Sliding Fit (Stationary Shaft)
Light	Up to and including 4" or 100 mm.	+ ·0002" to + ·0007"	+ ·0000" to - ·0005"
Medium or heavy	Up to and including 2" or 50 mm.		
Light	Over 4" or 100 mm.	+ ·0005" to + ·001"	- ·0002" to - ·0007"
Medium or heavy	Over 2" or 50 mm. and up to 4" or 100 mm.		
Medium or heavy	Over 4" or 100 mm.	+ ·0007" to + ·0012"	- ·0005" to - ·001"

TABLE II

OUTER RACES: BALL BEARINGS (JOURNAL TYPE)

Type of Bearing	Outside Diameter of Bearing	Bore of Housing	
		Interference Fit (Outer Rotating Race)	Sliding Fit (Stationary Outer Race)
Light or medium	Under 2"	- ·0005" to - 001"	+ 0000" to - ·0005"
	2" to 3"	- 0007" to - ·0012"	- ·0002" to - ·0007"
	3" to 5"	- ·001" to - ·0015"	- ·0005" to - ·001"
,	5" and over	- ·0015" to - 002"	- ·0005" to - ·001"

THE FIT OF THRUST BEARINGS

In the case of thrust bearings (Fig. 7), as the load on the balls is in the direction of the shaft axis, there is no load present to search for any clearance between the ball race and the part that it fits. It is therefore not necessary to make the races of ball thrust bearings a tight fit either on, or in, the revolving or the stationary member. As a general rule the revolving race is a push fit on, or in, the revolving member, whilst the fixed race is a push fit in its housing.

It is essential that the stationary housing be truly concentric with the rotating member's axis.

Further (Fig. 8), it is necessary to provide flanges, collars, or abutments for transmitting the thrust from the revolving member to its revolving thrust race, and from the other thrust race to its stationary housing.

Unlike journal type ball bearings, ball thrust bearings do not give rise to creep, as push fits are employed, but cases sometimes occur of thrust races creeping up against their abutments. If the stationary abutment is out of square with the axis of rotation, most of the load is transmitted to the revolving race at the place where the two races are closest together. This causes the point of maximum load on the revolving race to revolve as the race revolves. This, in turn, produces a revolving point of maximum load between the revolving race and its abutment, and " creep " then occurs. One should therefore look for the cause of " creep " in the non-alignment of the abutment of the opposite race. Another important point in connection with the fitting of thrust ball bearings is that the rotating member—the shaft, for example—should run perfectly true in its journal bearings. With plain bearings this is not always an easy matter to ensure, so that wherever possible journal ball or roller bearings should be used.

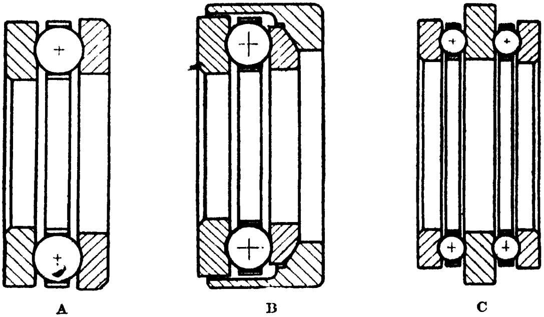

FIG. 7. EXAMPLES OF THRUST BALL BEARINGS

A = Single thrust type
B = Aligning single thrust type with outer housing
C = Double thrust type

When it is not possible to employ journal ball or roller bearings the thrust bearing must be protected from the effects of wear in the plain bearings, as this wear will cause inaccuracy of alignment. Fig. 9 shows a typical arrangement of a good method of protection. In this case both races of the thrust bearing are centred from the revolving shaft. If the plain journal bearing wears and allows the shaft to drop, both of the thrust races drop with the shaft and the tracks still remain concentric. It should be pointed out that the sleeve on which the stationary thrust race is mounted acts as a small journal bearing, taking the weight of the stationary race.

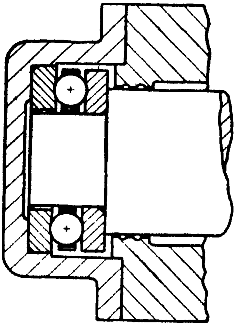

FIG. 8
SHOWING METHOD OF MOUNTING SINGLE THRUST BEARING

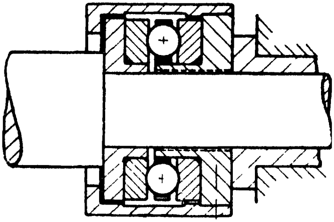

FIG. 9
A SINGLE THRUST BEARING MOUNTING FOR PLAIN JOURNAL BEARINGS

THE FIT OF DOUBLE THRUST BEARINGS

As we have already observed, the central ring of the ordinary double thrust bearing (Fig. 7 *C*) is thicker, and of larger diameter than the other two races. This central ring is usually clamped tightly to the fixed housing by means of a split cap or clamping ring provided with a recess to take the central ring near its outer periphery.

The other two outer races must rotate with the shaft, and at the same time be quite free from end play. These conditions necessitate the use either of a clamping nut and a shoulder on the rotating shaft or (what is much better practice, as there is less risk of clamping the balls too tightly against the races) of a separate screwed sleeve with its own shoulder or flange and nut. Such a unit can be supplied, properly adjusted and locked in position, ready for slipping over a shaft and clamping up against a suitable shoulder on the latter by means of a nut screwed on to the threaded part of the shaft (Fig. 10). The clamping pressure in this case is not transmitted through the balls and their races, but only through the material of the sleeve. This is no doubt the better method, as it relies upon the ball bearing manufacturer adjusting the thrust races to the correct clearances, and also as it relieves the thrust bearing of only clamping stresses.

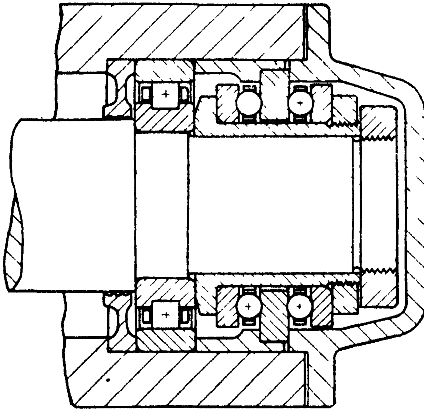

FIG. 10. A TYPICAL DOUBLE THRUST BEARING MOUNTING

In the example shown in Fig. 10 the sleeve carrying the double thrust bearing abuts against one face of a journal roller bearing, taking the journal load at this end of the rotating shaft. The complete thrust unit (with its sleeve) is held against the face of the inner ring of the journal bearing by means of the clamping nut shown at the end of the shaft. The central ring member of the thrust race is clamped securely and centrally to the stationary, or outer, housing by means of the cap shown. The latter has a register boss or ring to enable it to enter the right-hand portion of the housing quite centrally. This "locating" ring method of ensuring centrality of a fixing member is widely used in ball bearing design. It should be noticed that the clamping cap does not bed flush with its fixed member on the right, but that a small "clamping" clearance is left.

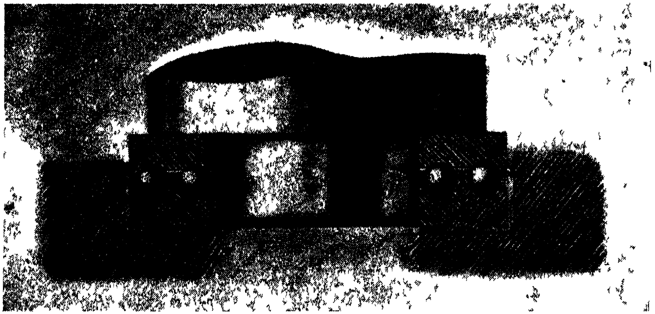

FIG. 11. THE HOFFMANN DOUBLE ROW, HEAVY DUTY THRUST BEARING

DOUBLE ROW, HEAVY DUTY THRUST BEARINGS

Where particularly heavy end loads or thrusts come upon rotating shafts, as in the case of pedestals, turntables, and crane posts, it is usual to employ a special type of thrust race having two rows of balls, as shown in Fig. 11. In this example the moving race has two concentric grooves for the balls, whilst the fixed or lower race is made in two separate parts that rest upon a ring having annular faces on the top side and flat at the bottom. This ring is purposely made elastic by means of a large number of saw cuts made radially in opposite directions and extending almost entirely through the section. The thrust from each row of balls is carried directly on this ring, and the load is thus distributed uniformly.

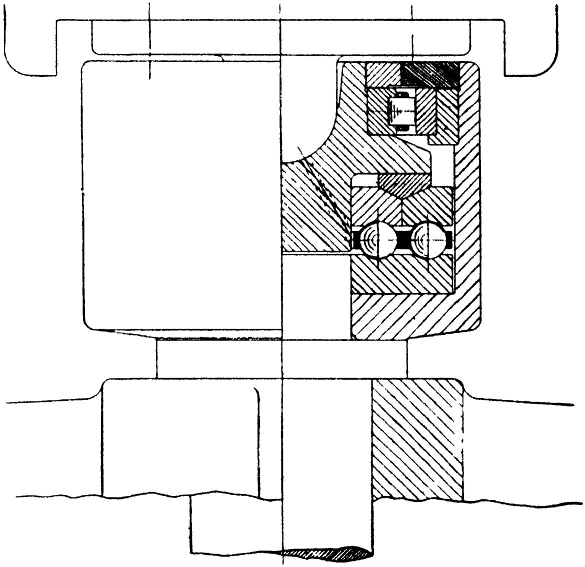

FIG. 12. SHOWING TYPICAL APPLICATION OF DOUBLE ROW THRUST BEARING

Fig. 12 illustrates a typical application of a Hoffmann bearing, of the double row variety, to a vertical shaft.

CUP-AND-CONE BEARINGS

In many cases, more particularly where lightly-loaded bearings are concerned, there is not room to fit a single or double thrust ball bearing; moreover, the extra weight of thrust or double taper roller bearings is not always permissible in many cases. It is therefore necessary to make some form of compromise by combining the journal ball bearing and thrust ball bearing in a single unit. Fig. 13 illustrates a bearing of this class, designed to carry both a journal load and to take end thrust in one direction. The bearing shown is known as the *cup-and-cone* type, the inner member being termed the *cone*, and the outer one the *cup*. It gives a two-point contact for the balls, and the radii of the races are rather greater than those of the balls. As a general rule, the radius of each curved race (shown at *R*, Fig. 13) is about 10 to 15 per cent greater than that of the balls, i.e. about $1\frac{1}{8}$ times the ball radius.

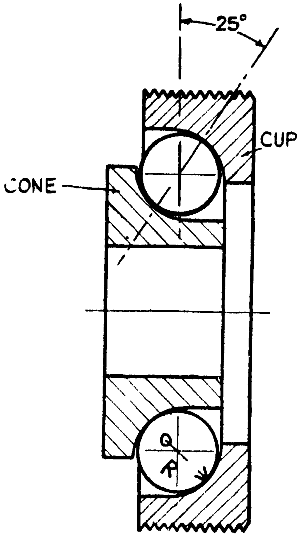

FIG. 13. THE CUP-AND-CONE BEARING

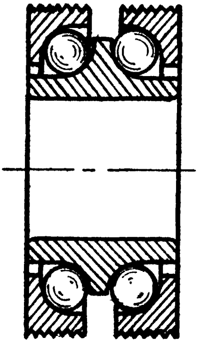

FIG. 14. THIS BEARING TAKES THRUST IN BOTH DIRECTIONS

The points of contact of the balls and their races should lie on a line passing through the centre of a ball and inclined to the axis at 25 deg.

As the engineer may have to make this type of journal thrust bearing, it may be mentioned that the races are usually made of a case-hardened nickel steel, or a high carbon steel tempered to straw colour and then finished by grinding. There is no cage for the balls in the case of pram and cycle bearings, and it is usual to insert a sufficient number of balls just to fill the space between the cup and cone, allowing a small amount of clearance between the balls to reduce the rubbing tendency of individual balls.

One or both of the races is threaded with a fine-pitch screw thread cut in the lathe, dead true with the cone or cup. It is important to provide adequate means for locking the races on their shafts or in their housings.

In the case of cycle-type bearings the cones are usually turned solid with the fixed wheel spindle, and the cups are threaded externally and screw into the wheel (or pedal) hub. A screwed locking ring is employed to lock the position of the hub.

In connection with the use of cup-and-cone bearings it is usual to fit these in opposing pairs, so as to take thrust in both directions, as shown in Fig. 14.

DOUBLE PURPOSE BALL BEARINGS

The cup-and-cone ball bearing is now seldom used, except for cycles and certain small machines and instruments, its place having been taken by special designs of dual purpose bearings marketed as standard self-contained components of compact form and easily assembled. Fig. 15* shows a typical dual purpose bearing for taking both journal, or radial, and thrust loads. The lower race resembles that of the deep race plain journal bearing, whilst the upper one is radiused, as shown, to take the thrust as well as load. the resultant line of action is indicated by the oblique line through the centre of the ball. This bearing is made so that it can be handled as a single unit and has a one-piece cage without rivet holes, so that a greater number of balls can be employed than in the usual cage-type

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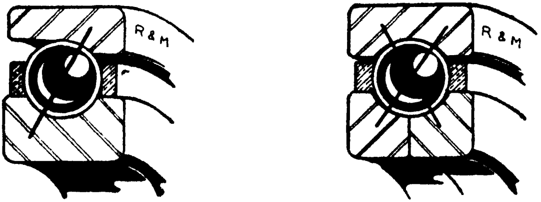
* Courtesy *The Mechanical World.*

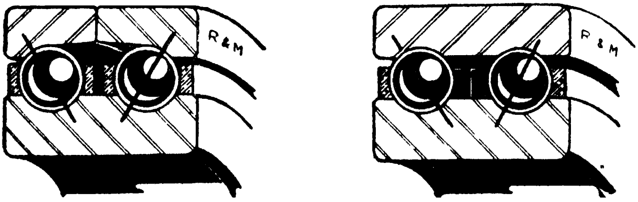


Figs. 15 and 16 Examples of Dual Purpose Thrust and Load Bearings

ball bearing. It is, of course, necessary to adjust the bearing in its position on the machine or other part to which it is fitted, so as to ensure correct contact and tracking of the balls and their races.

It is usual, where thrusts have to be taken in both directions to fit a pair of these bearings with the thrust faces facing one another. In cases where there is very limited space for ball bearings to take both load and end thrusts in either direction a more suitable type of bearing is that shown in Fig. 16. This bearing, which is made by Messrs. Ransome & Marles Company, has a solid outer race and two symmetrical inner races; alternatively it can be made with a solid inner race and a split outer one. This design of bearing is intended, primarily, as a double thrust one, but it will take a certain journal load, namely, about one-quarter of the total thrust load and therefore may be employed as a dual purpose thrust and load bearing.

In instances where greater journal loads have to be taken, with provision for taking thrust in either direction, the designs of dual purpose bearings shown in Fig. 17 (*a*) and (*b*) should be used. The former type may be regarded as being equivalent to a pair of opposed bearings of the kind shown in Fig. 15, but arranged as a self-contained unit, i.e. with a single inner race. It is necessary for this bearing to be mounted with the abutting faces on either side of the outer races so that it should not be used for purely journal loads; this type of bearing is made in such a way that it does not require any adjustment when it is fitted on its shaft. The bearing shown in Fig. 17 (*b*) has solid inner and outer races, and it will take full journal loads as well as thrust ones. It employs one piece cages, and since no filling gaps are used for the insertion of the balls, a smaller number of balls is used than for the bearing shown in Fig. 17 (*a*); in consequence the load capacity is reduced to some extent.



(*a*) (*b*)

Fig. 17

Advantages of the dual purpose bearings described are their compactness, ease of fitting, and relatively lower cost than when separate journal and thrust bearings are employed. Further, when subjected to end loads at high speeds, the balls are restrained under centrifugal loading, and these bearings are therefore much less liable to fail at extreme speeds than other types having balls constrained between pairs of washers.

An illustration of the application of a dual purpose bearing is given in Fig. 18, which shows a Delco Hyatt "Radax" ball bearing fitted to one end of a shaft. The angular line of contact is 15 deg. The housing for the outer race is provided with a screw cap to seal the bearing against entry of dust or water, and at the same time to provide a reservoir for the bearing lubricant.

A special type of light ball bearing is supplied by most ball bearing manufacturers for use on the armature shafts of magnetos and on battery ignition contact-breaker shafts. These bearings take only light loads, and are used in cases similar to that of the magneto where the inner race has to be quickly and readily removed from the outer race and housing. They are also used for very high speeds, such as for polishing buffs and grinding spindles.

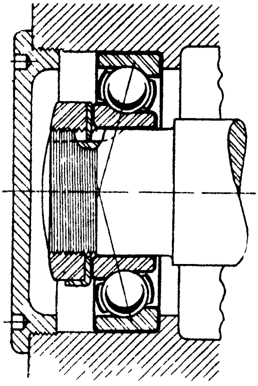


FIG. 18. THE "RADAX" COMBINED JOURNAL AND THRUST BEARING

Magneto type bearings are supplied to suit shaft sizes of 5 mm. up to 19 mm. (in millimetre steps in the case of Hoffmann bearings). These bearings, in order to locate the spindle on which they are mounted, must be fixed in pairs, each bearing preventing end movement in one direction. It is important to ensure that the axial adjustment is not so tight as to cause overload on the balls and races.

The adjusting feature in this case is provided merely to secure correct fitting of the parts in the first instance, and not as a means of taking up wear.

The three components of magneto type ball bearings, namely, the two races and the cage with its balls, are usually separable to facilitate fitting of the bearing.

SELF-ALIGNING JOURNAL BEARINGS

Where more than one ball bearing is used on a shaft it is not always possible to align the bearings correctly, and in cases in which the shaft itself may flex or move slightly in axial direction there would be a big stress on the bearings, in the ordinary way, if plain single row journal ball bearings, rigidly mounted as recommended in ball bearing practice, were employed. These bearings would be subjected to undue wearing action, for which they were not designed.

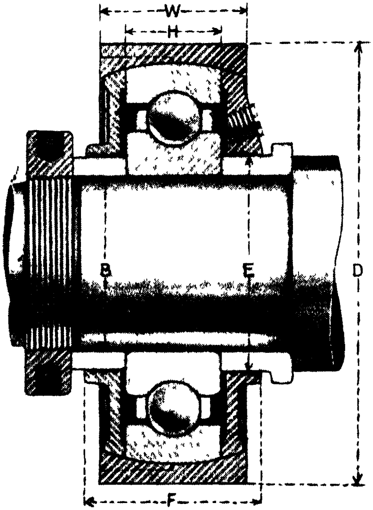


FIG. 19. THE HOFFMANN SELF-ALIGNING JOURNAL BALL BEARING

It is necessary, therefore, to employ a special type of ball bearing such that whilst capable of carrying its journal load and acting as a journal bearing, it is able to adjust or to accommodate itself to deflection or misalignment of the shaft to which it is fitted. Two different methods of obtaining this accommodation or self-alignment are illustrated in Figs. 19 and 20.

Fig. 19 shows the Hoffmann self-aligning bearing. In this case the inner race is secured firmly to the shaft, whilst the outer race has a spherical surface such that it can slide in a spherical seating in another housing member. In the example shown, the inner race is clamped securely between two sleeves, the right-hand member of which bears against a shoulder on the shaft. The screwed locking ring on the left clamps the two sleeves together. A dust cover, with grooves in its inner edge, is provided on each side of the two ball races, to keep out dust and water, and to retain the grease injected through the screw hole shown in the right-hand member. Another design of this self-aligning bearing has a split tapered sleeve for clamping securely on to parallel shafts, so that the shoulder in the previous example is not required. These bearings are designed for journal loads only, and it is necessary that the outer member be free endways to allow it to take up a position opposite the fixed inner race. The shell or housing should accordingly be a nice push fit, free from shake, but not so tight as to prevent the balls moving axially if required. The clearance on either side of this shell should be about one-third of the total width of the bearing.

In the case of the Skefko self-aligning ball bearing (Fig. 20) the inner surface of the outer ball race is made of spherical form, with the centre of the sphere on the axis of the shaft. When the latter flexes or distorts, the balls simply take up a new position in their spherical outer race, and thus in a frictionless and perfectly simple manner accommodate themselves to any change of direction. As the correction for non-alignment is allowed for in the bearing itself, these bearings require no special external members, are self-contained, and can be made to approximately the external dimensions of ordinary journal ball bearings. In effect, we have in this type of bearing the ball-and-socket principle. It will be observed that there are two rows of balls, cleverly spaced by means of a sheet metal cage.

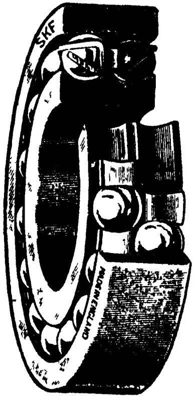


FIG. 20. THE SKEFKO SELF-ALIGNING BALL BEARING

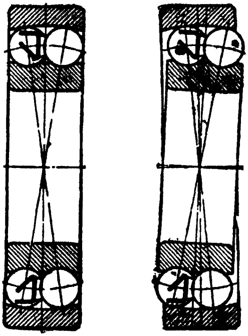


FIG. 21. SHOWING THE PRINCIPLE OF THE SKEFKO SELF-ALIGNING BALL BEARING

Self-aligning journal bearings are particularly applicable to line shafting of engineering shops and industrial factories, to the bearings of overhung shafts, drilling machine spindles, grinding and polishing machines, for electrical machines, woodworking machinery, certain automobile parts, textile and paper-making machines, marine work, printing machines, agricultural and mining machinery, and in many other miscellaneous spheres.

ROLLER BEARINGS

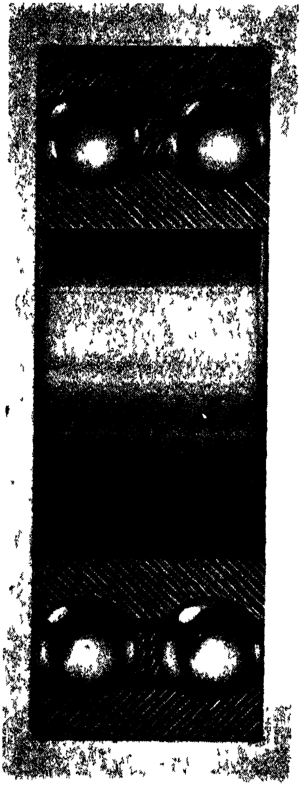


Fig. 22. The Hoffmann Double Row Bearing for Heavy Loads

As we have stated, the ball bearing works on the principle of point contact of the balls on their race grooves, and each size of bearing will only safely work with a given maximum load. Where heavier loads and shocks have to be provided for, there are two alternatives to the single row journal ball bearing, as follows. (1) The Double Row Ball Bearing (Figs. 20 and 22), or (2) The Roller Bearing. The latter type gives line contact to the rollers on their races, and so is able to bear much heavier loads than the single or double ball type journal bearings. In the past the design of roller bearings has been attended with greater difficulties than in the case of ball bearings, for troubles were experienced in connection with keeping the axes of rotation, or rolling of the rollers, strictly parallel with the shaft axis, and in preventing undue end thrust of the rollers.

Roller bearings are now available in two principal types, namely, the plain parallel roller journal bearing and the tapered roller bearing. The former is usually unable to deal with end thrust, whereas the latter can take thrust in one or both directions in addition to journal loads. Roller bearings are made also in the *self-aligning* as well as the plain journal, or journal thrust types.

Fig. 23 illustrates the Hoffmann plain roller journal bearing, in which, in place of balls, there is a row of short rollers each of length equal to its diameter. This type will carry a load from 50 to 70 per cent greater than a ball bearing of the same dimensions. It will not, however, take any end thrust so that some means must be provided for end location, either by plain thrust washers—in the case of light thrusts—or ball thrust bearings. In many cases the manufacturers recommend the use of a "*location*" ball bearing of the journal type; this is similar to a standard ball bearing, but slightly smaller in outside diameter. It is mounted immediately alongside the roller bearing, the housing being bored straight through to suit the outer race.

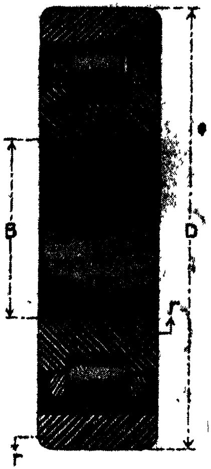


Fig. 23. The Hoffmann Plain Roller Bearing

SOME IMPORTANT MOUNTING PRINCIPLES

To obtain the most satisfactory service from ball and roller bearings these must be mounted correctly in relation to the fixed and moving members. A few typical illustrations will at once demonstrate the principles to be followed in designing the housings for these bearings.

Taking the simplest case met with in engineering workshop practice first, namely, that of a rotating shaft supported by and running in two fixed journal ball bearings (Fig. 24, *A*), the mountings must be accurately aligned, so that their axes are coincident. One of the bearings is used to hold, position, or locate the shaft; in this case it is the left-hand bearing. The fixed outer race in this case is a sliding fit in the housing, and it is clamped securely on both sides by the housing, so that it cannot move in an axial direction.

In this method of mounting, as some allowance must be made for end play and expansion, it is usual to allow the outer race of one bearing (in this case the right-hand one) freedom to slide a little each way in its housing; the method of accomplishing this will be clear from Fig. 24, *A*, (right-hand member). The inner races in both cases are interference fits on the shaft, whilst the outer races are sliding fits.

Fig. 24, *B*, shows the same example of a two-bearing shaft, but with a roller bearing in place of the right-hand ball bearing; this enables it to take a greater journal load at this end. As the rollers are not constrained in any way on the outer cylindrical race, there is no need to allow for any end play in this member as in *A*, Fig. 24.

In connection with the clamping of the inner races it should be noted that the clamping surfaces do not abut against the sides of the roller track to a greater diameter than the bottom of the roller track, the clamping nuts being chamfered for this purpose. If the shaft of the two previous examples is subjected to a side thrust in either direction, and this thrust is too great to be taken by the journal ball bearing, a thrust washer should be arranged in the housing of the fixed or locating end of the shaft.

Fig. 24, *C*, shows how this is carried out. It will be

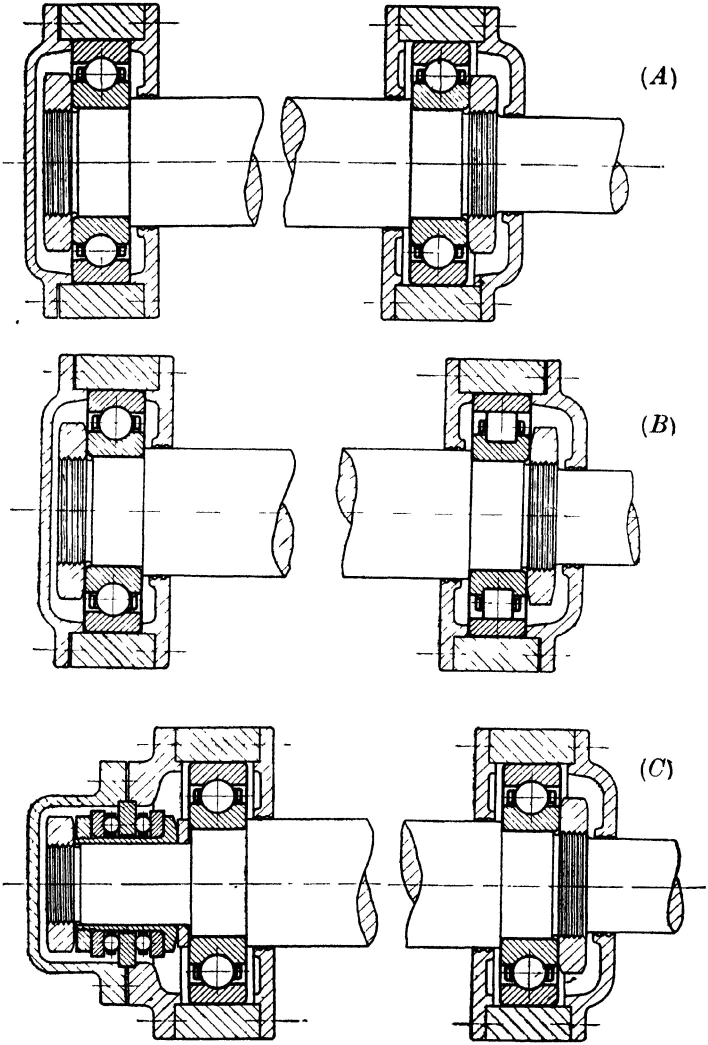


FIG. 24. SOME TYPICAL METHODS OF MOUNTING SHAFTS IN BALL BEARINGS

noticed that since the double thrust bearing now locates the shaft, the left as well as the right ball bearing outer race is allowed end play in its housing. The Hoffmann thrust bearing shown is a complete unit on a sleeve, correctly adjusted for use by the manufacturers; the sleeve is a push fit on the shaft. If instead of journal ball bearings a pair of roller bearings has to be used on account of heavier journal loads, there is no need to allow any freedom for end movement of the outer races, so that these may be fixed endwise in their housings; any end movement that may be necessitated is allowed for by the sliding freedom of the rollers in their outer cylindrical races.

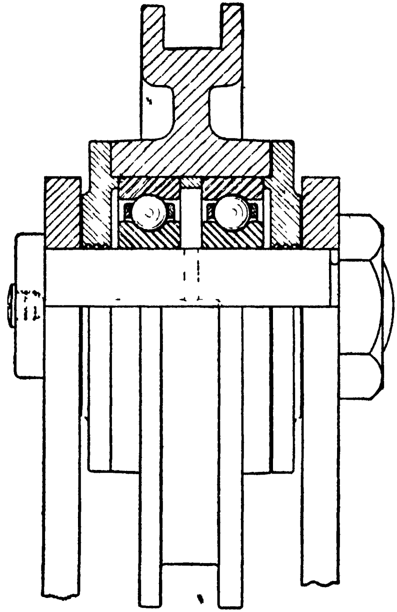


FIG. 25. APPLICATION OF BALL BEARING TO A FLANGED PULLEY OR SHEAVE

Another typical application of ball bearings is illustrated in Fig. 25, namely, that of a flanged pulley running on a fixed shaft. In this case the outer races rotate, and are therefore clamped, with a suitable distance piece between, solid inside the pulley. The inner races are a sliding fit on the pin or shaft, whilst the outer races are an interference fit in the pulley. The pulley is located by arranging a running clearance

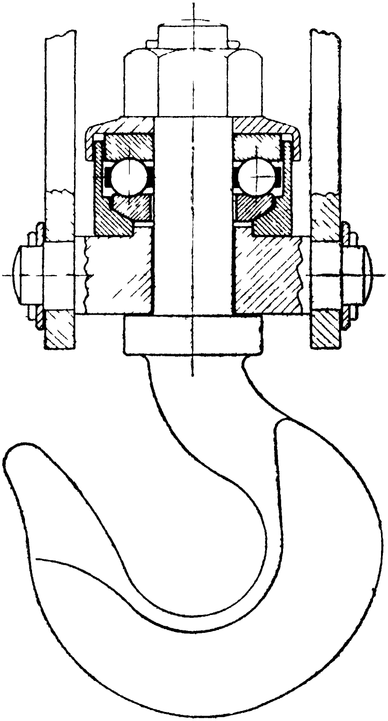


FIG. 26. EXAMPLE OF THRUST BEARING APPLIED TO A CRANE HOOK

between the external faces of the end covers and the side plates shown. The method of protecting the grease-packed bearings with the close-fitting grooved end cover plates should be noted in this example. Fig. 26 illustrates a good example of the use of a ball thrust bearing (Hoffmann) in the case of a large crane hook. A somewhat similar arrangement is used for wire rope shackles and deep-well boring. In this case the shank of the hook is made with a running clearance in the bridge piece, and concentric with the bore of the latter should be a spigot centring the housing of the thrust bearing. The lower face of the nut should run true;

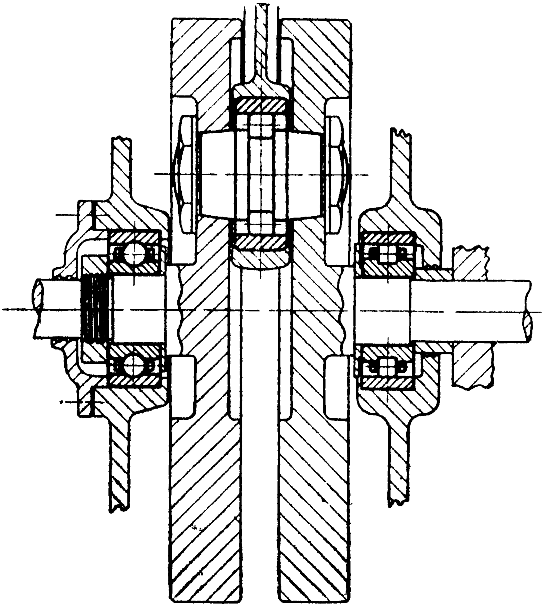


FIG. 27. SHOWING BALL AND ROLLER BEARINGS APPLIED TO MOTOR-CYCLE ENGINE

between it and the top race of the thrust bearing is a dished cover to keep out dirt and water. It will be noticed that the nut is tightened on its thread until the shoulder of the shank is just clear of the lower surface of the bridge piece. The housing is filled with grease. Figs. 27 and 28 show how ball and roller bearings are applied to motor-cycle type engines.

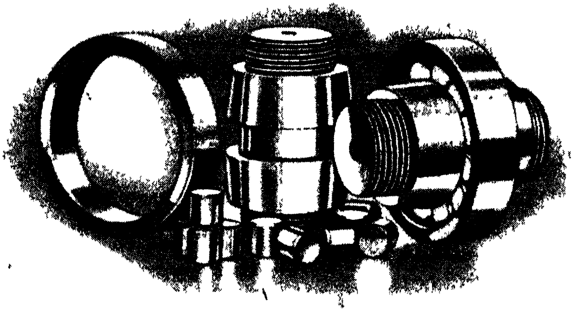


FIG. 28. ROLLER BEARINGS APPLIED TO BIG END OF MOTOR CYCLE CONNECTING ROD

MINIATURE AND LIGHT LOAD BALL BEARINGS

The ball bearings used in small electric motors, meters, clockwork and pressure gauges, light machines and instruments are made in small sizes ranging from 1½ mm. upwards. For small electric motors, fans and similar parts the bearings are made for shafts of ¼ in. to about ½ in. diameter.

The miniature bearings shown in Figs. 29 and 30* are of the pivot pattern for use with a conical end shaft so that they will take journal and thrust loads in one direction. These small bearings* actually give a much lower frictional drag than jewelled bearings such as ruby and sapphire ones, and they exhibit almost the same frictional value when starting as when running; this is not the case with plain bearings which have increased starting friction.

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* *The Engineer.*

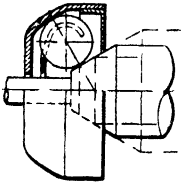

FIG. 29

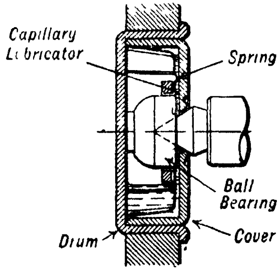

FIG. 30

The illustration in Fig. 30 shows a typical application giving a certain amount of flexibility in order to withstand small shocks or impacts. In this arrangement a drum encloses the bearing and a spring; the latter fulfils the double function of shock absorber and lubricant feeder through a small channel formed inside the drum. The complete unit is closed by a cover which is pressed into position. The effect of shocks is to throw the spindle out of position at the time, but it returns at once to its normal operating position.

In connection with the subject of miniature ball bearings, what is claimed to be the smallest commercial bearing is that made by the Hoffmann Manufacturing Co. Ltd. It is of the journal type and uses balls of $\frac{1}{64}$ in. diameter made to a high standard of precision.

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* International Technical Developments, Ltd., Thames House, Millbank, London, S.W.1.

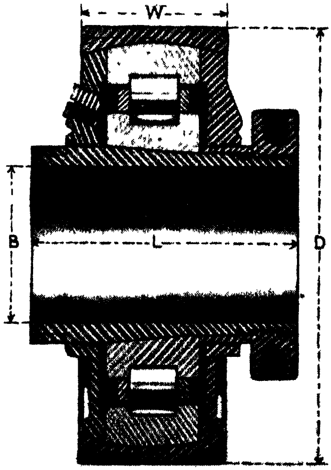

FIG. 31. THE HOFFMANN SELF-ALIGNING ROLLER BEARING

This bearing embodies the same principle as that of the self-aligning ball bearing (see Fig. 19)

SELF-ALIGNING ROLLER BEARINGS

Fig. 31 illustrates the Hoffmann self-aligning roller bearing which is similar in principle to the Hoffmann self-aligning ball bearing.

The Skefko single row cylindrical roller bearing is illustrated in Fig. 32, whilst the self-aligning type is shown in Fig. 33. It will be seen that there are two rows of curvilinear short rollers, which roll on the spherical outer race in such a manner that they can adapt themselves to flexure of the shaft, as in the

Skefko self-aligning ball bearing. The Skefko self-aligning roller bearing may be regarded as a heavy duty bearing with self-adjusting properties, capable of withstanding working loads of about twice the value of

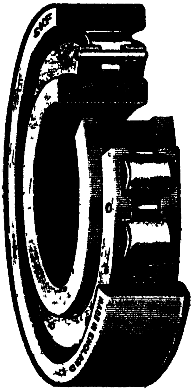

FIG. 32. THE SKEFKO SINGLE ROW ROLLER BEARING

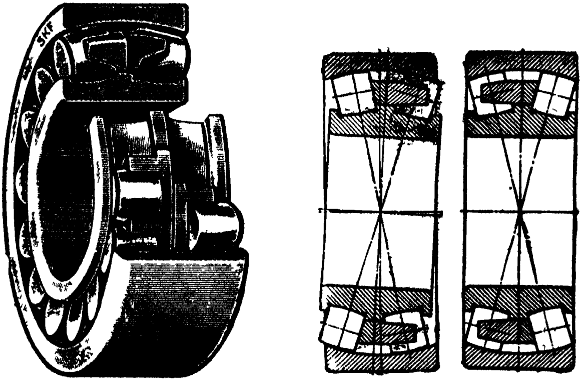

FIG. 33. THE SKEFKO DOUBLE ROW ROLLER SELF-ALIGNING BEARING

those carried by ball bearings of the self-aligning type. These bearings are supplied to suit shafts varying from 50 mm. (about 2 in.) up to 200 mm. (about 7·9 in.).

Where the journal roller bearings have to withstand also an end thrust in one direction only, it is usual to employ special types of roller bearing having tapered rollers and conical surfaced races for the rollers to run upon.

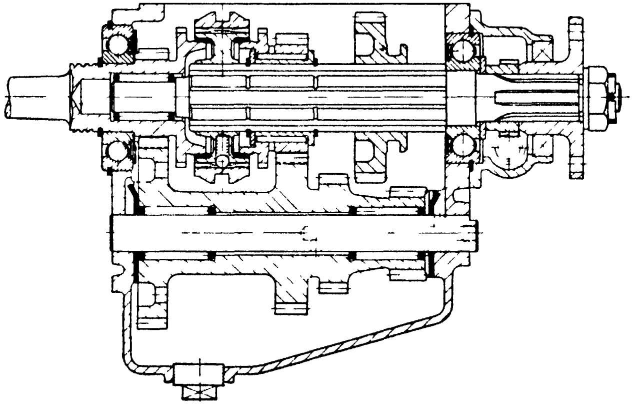

FIG. 34 AUTOMOBILE GEAR-BOX

APPLICATIONS OF BALL AND ROLLER BEARINGS

Some typical applications of Skefko ball and roller bearings to automobile engineering are given in Figs. 34, 35, and 36.

Fig. 34 shows a sectional view of a motor-car gear-box the bearings of which have to carry heavy radial loads and considerable end thrust. On account of space restrictions a bearing of small outside diameter is necessary for the spigot position. With the helical gears constantly in mesh it is usually necessary to provide a bearing between the gear and the shaft which will function when the gear and shaft speeds differ. For the constant mesh pinion and mainshaft rear end single row deep groove bearings are used; these can be provided with spring rings on the outside diameter for location purposes; they can also be fitted with oil shields. A solid roller assembly is used for the spigot and the same type may be used on a hardened layshaft when the gears are of the cluster type. For the second speed helical wheel needle rollers* are provided; these run on a hardened sleeve fitted over the mainshaft and located laterally by means of spring clips or "Circlips."

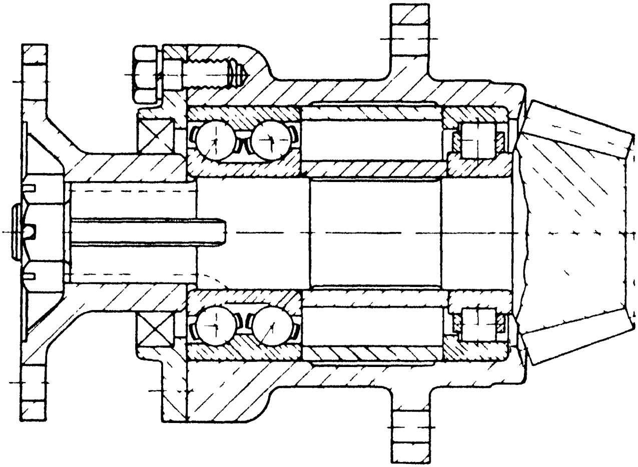

FIG. 35 AUTOMOBILE BEVEL PINION BEARINGS

Fig. 35 shows a well-designed bevel pinion bearing for the final drive, i.e. between the rear end of the propeller shaft and the crown wheel of the back axle. The bearing immediately behind the pinion is a roller one, whilst that carrying the thrust and remaining radial loads is a double-row ball bearing with opposed angular contact—to take end loads in both directions.

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* See page 117.

Fig. 36 shows a commercial vehicle steering gear-box of the worm and sector pattern. The steering column is attached to the worm member, and the latter is provided with a pair of opposed single race dual purpose bearings to take both journal and end loads. The outer rings are pressed into the end caps whilst the inner rings are made a tight fit on the worm-shaft. Endwise adjustment is provided by means of shims fitted between the bottom cover and the casing.

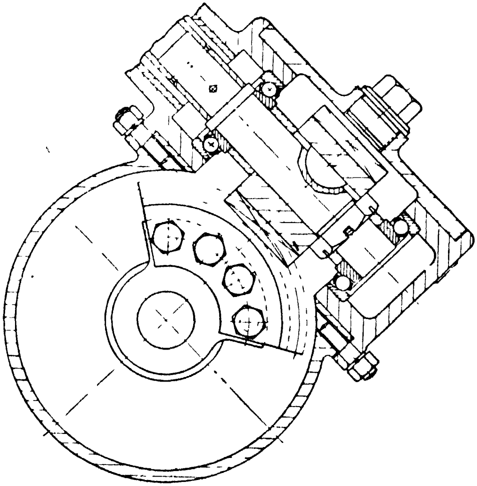


FIG. 36. STEERING GEAR-BOX BEARINGS

THE TIMKEN ROLLER BEARING

Figs. 37, 38, and 39 illustrate what is probably the best-known and most widely used taper roller bearing, namely, the Timken. In this case there is a series of hardened steel tapered rollers, spaced apart by means of a specially designed cage, having roll pockets punched by a process that obviates distortion. The cage ribs are winged so that they conform to the curvature and taper of the roll. The Timken cage thus provides a pocketing for the rolls which is accurate, thus allowing each of the rollers, without interference, to perform an exactly equal amount of work.

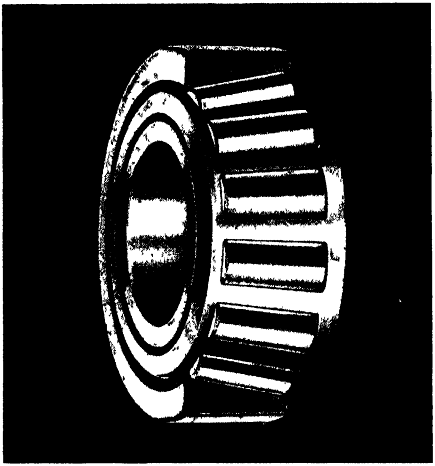


FIG. 37. THE TIMKEN TAPER ROLLER BEARING

The principle of the Timken taper roller bearing is illustrated clearly in Fig. 39. It will be observed that the races and rollers are all portions of conical surfaces, the apices or conical points meeting at a fixed point on the axis of the shaft to which the bearing is fitted. This is the essential condition for pure rolling to be obtained without sliding in any tapered roller type of bearing.

In reference to this type, it is of interest to mention that the *inner race* is termed the *cone*, and the *outer race* the *cup*. The shape of the cone should be noted carefully, for it has a raised rim or rib at each side, the object of which is to provide normal contact areas for the rollers. The roller has at its larger diameter end a flat surface which operates against the rib of the cone. This flat surface at the end of the roller is made to square exactly with the centre line of the roller itself. As each roller revolves about the cone, a normal contact is made against the cone rib in two areas. This contact compels the revolving roller to align itself correctly under all running conditions, and ensures that a full-line contact is maintained constantly between the roller and the cup on the one hand, and the roller and the cone on the other.

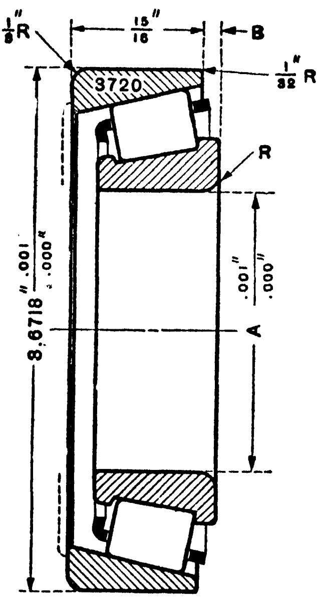


FIG. 38. A TYPICAL TIMKEN BEARING (SHOWN TO SCALE)

Timken bearings are made in a wide variety of sizes and load capacities for numerous purposes. The sizes vary from those small enough for sewing machines, lawn mowers, and small electric fans, up to large ones employed to withstand exceedingly high thrusts and radial loads, as in steel rolling mills. Between these come the many intermediate sizes found in line shaft hangers, lathes, drills, presses, grinders, planers, roll conveyors, hoisting pulleys, and numerous applications in the motor-car, commercial vehicle, and tractor. Another more recent application is in connection with railway carriage axle bearings, to replace the ordinary plain metal bearings, for tests have shown convincingly that a considerable reduction in axle friction is obtained with roller bearings.

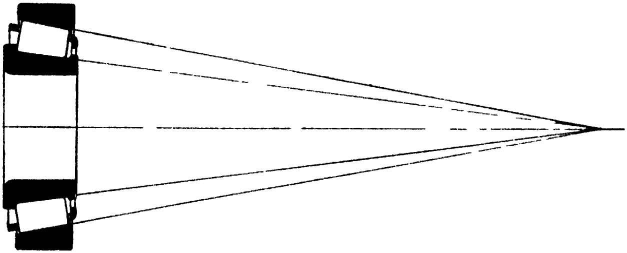


FIG. 39. SHOWING THE PRINCIPLE OF THE TIMKEN ROLLER BEARING

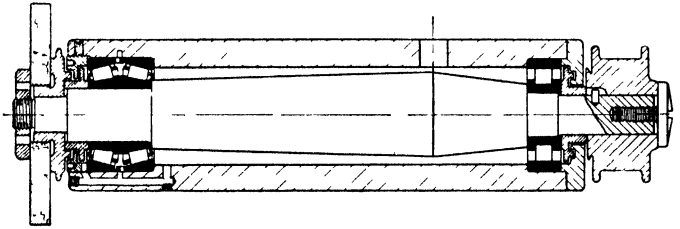


FIG. 40 GRINDING SPINDLE BEARINGS

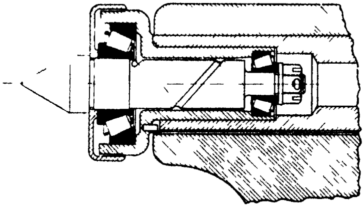


FIG. 41 LATHE TAILSTOCK BEARINGS

Figs. 40, 41 and 42 show some typical applications

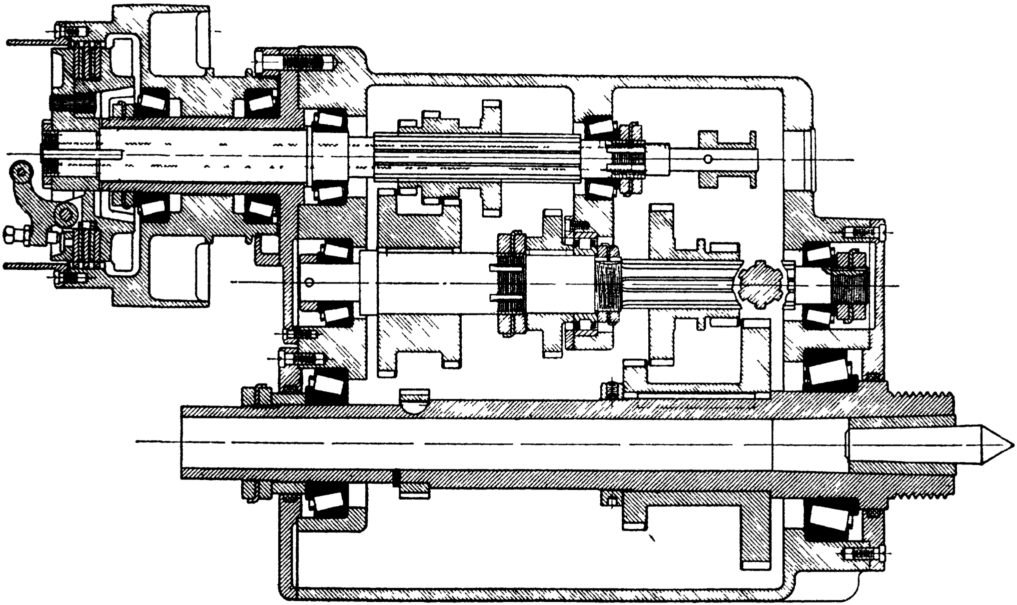


FIG 42 LATHE GEARED HEADSTOCK BEARINGS

of Timken tapered roller bearings to machine tools. Fig. 40 illustrates the use of these bearings for a high speed internal grinding spindle. The two tapered roller bearings next to the wheel are located axially in a positive manner; adjustment is obtained by means of the spacer ring between the cups and the finely threaded cup follower. The parallel roller bearing mounted adjacent to the pulley enables the spindle to float axially so as to allow for any heat expansion effects.

Fig. 41 shows a roller bearing tailstock centre for a small lathe, with provision for taking up end and radial wear by means of a single adjustment, namely, the castle nut on the right.

Fig. 42 illustrates the geared headstock of a modern lathe and indicates clearly the application of tapered roller bearings for the various shafts, the lathe spindle (below), and the driving pulley bearings.

THE MOUNTING OF TAPER ROLLER BEARINGS

The following information relates to the mounting of Timken bearings of the type just described. It will be evident from what has been stated that the best disposition is to arrange the bearings in pairs, either with the conical surfaces converging inwards, as shown in Fig. 44 (above), and known as the "*indirect*" method, or outwards, as in Fig. 44 (below), and known as the "*direct*" method. The former method is practically standard in all stationary shaft applications. A good example is that of the front axle of automobiles, the stub axles housing the fixed races or cones, and the moving wheel hubs the cups. Other examples of indirect mountings include farm implement wheels, conveyor idler pulleys, mine car wheels, overhead trolley wheels, logging blocks, sheave wheels, hoists.

drums, crane wheels, and all similar dead shaft applications. Experience has proved this type of mounting superior to the direct mounting one for wheel or pulley service.

In indirect mounting the distance between the bearing centres should be not less than 15 per cent of wheel outside diameter, and preferably 20 per cent.

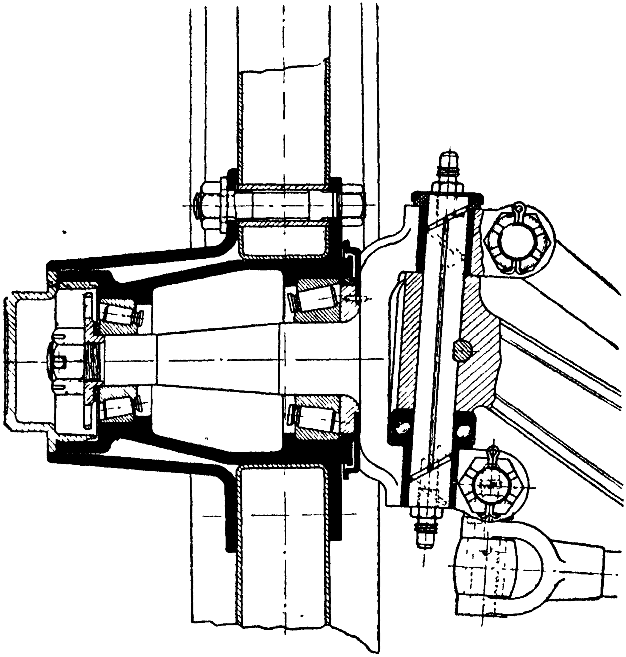
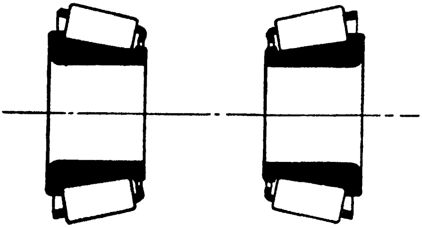
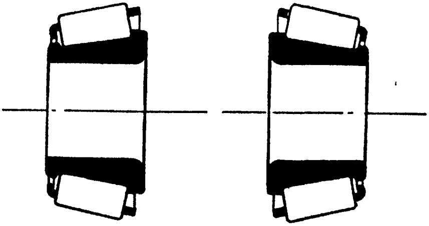


FIG. 43. AUTOMOBILE FRONT AXLE BEARINGS

Adjustable bearing cones with light fits are generally used with the indirect mounting on stationary shafts for speeds below 1000 rev. per min. Slight cone creepage, which results from light fits, is quite permissible on stationary shafts at these speeds. Above 1000 rev. per min. the indirect mounting and adjustable cones are used with tighter cone fits ; in this case cone creepage is undesirable, and must be guarded against by tighter fits.



INDIRECT



DIRECT

FIG. 44. METHODS OF MOUNTING ROLLER BEARINGS

The manufacturers give tabulated particulars of the sizes, loads, speeds, and fits of their different bearings.

THE DIRECT SYSTEM OF MOUNTING

The direct system of mounting (Fig. 44) is used in applications with relatively long bearing centres. It is more satisfactory above 1000 rev. per min. than the indirect mounting, because the cones can be given the tight press fits required at these speeds. This method is used widely on automobile transmission gear shafts, differential gear shafts, and live axles as well as machine tool transmission shafts, for general gear reductions, pillow blocks, journal boxes, and other applications which permit large bearing spans.

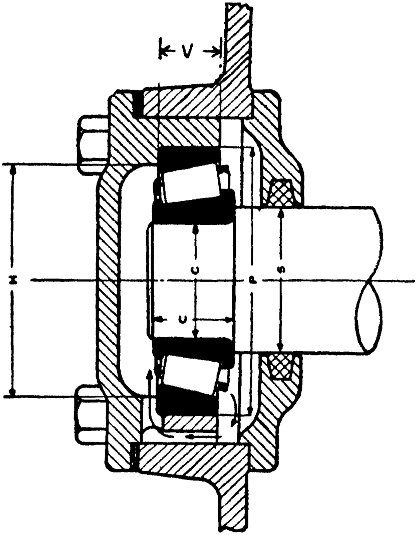


Fig. 45. Illustrating Machining Dimensions of Parts for Taking Timken Bearings

MACHINING PARTS FOR TAKING ROLLER BEARINGS

Referring to Fig. 45, which shows the principal dimensions of the shaft and housing for Timken roller bearings, it is very important to ensure that the machine work is carried out within specified limits to obtain the best working conditions. In regard to the "width" dimensions, U and V (Fig. 45), width tolerances of ·020 in. to ·030 in. can be permitted in the machine work at each bearing by making provision for this variation in bearing adjustment.

Machine work for two opposed bearings may have total tolerances of ·040 in. to ·060 in. and be speeded up by the use of machine stops to gauge shaft shoulder width measurements.

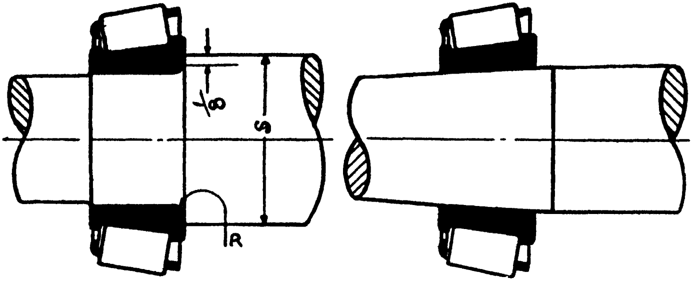


FIG. 46. TWO METHODS OF CONE MOUNTING

A shaft diameter *C* (Fig. 45) with a tolerance dimension varying from ·0006 in. to ·0015 in. according to bearing bore is standard practice for cone seats. The variation allowable in the housing bore dimension *P* is from ·001 in. to ·002 in. for non-adjusting cups with tight press fits, and from ·0006 in. to ·0014 in. for adjusting cups with light press fits. It is absolutely essential to ensure correct alignment of the cone and cup seats.

Where possible, the cone should be located on its shaft by means of a shoulder turned on the shaft $\frac{1}{8}$ in. higher than the radius *R* (Fig. 46); the fillet radius should be $\frac{1}{64}$ in. less than the cone radius. The shaft diameter *S* for all standard cones should be estimated from the dimensions given in the maker's catalogues.

In some cases tapered shafts (Fig. 46) are employed for the cone housing. In these instances special tapered bore cones are obtainable. This type of fixing eliminates shoulders, and is used on live rear axles of automobiles.

ADJUSTMENT OF TIMKEN BEARINGS

As we have mentioned previously, adjustment of this type of bearing is a relatively easy matter, and is accomplished by moving either the cup or cone (whichever is the stationary member) in an axial direction. This feature enables wider tolerances for machining bearing seats to be employed, and makes possible the adjustment of other dependent parts, such as spiral bevel gears, helical gears, and the front axle bearings of automobiles. Further, it is possible by this adjustment to eliminate chatter on machine tool spindles, and generally to reduce noise.

General applications should have ·002 in. to ·006 in. end play. End movement of either the stationary or the revolving race, properly fitted, can be used to adjust the bearings for speeds up to 1000 rev. per min. Above 1000 rev. per min. it is considered advisable to make adjustments with the stationary race in order to permit a tight press fit of the rotating race. It is recommended that the adjustment increments should not be greater than ·005 in. In the case of two opposed bearings a total end movement of $\frac{1}{32}$ in. should be arranged for. This amount should allow for and include the variations due to machining tolerances, variations in bearing widths, and subsequent adjustments of the bearings. *Threaded adjusting nuts* are widely used for either the cup or cone movement. *Light or medium duty* applications should use nuts having *sixteen threads per in.*, and six or more locking castellations. *Heavy duty* machinery may require nuts with *twelve threads per in.* and eight castellations. The threads should be close fitting. Split nuts or housings that can be clamped on mating threads are essential on *extra heavy jobs.* Threaded adjustments should not only be positively locked against rotation, but, wherever possible, the threads should be clamped *to ensure tightness against end play.*

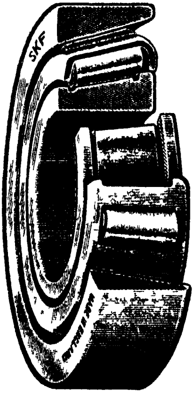


FIG. 47
THE SKEFKO
TAPER ROLLER
BEARING

SOME OTHER ROLLER BEARINGS

The Skefko taper roller bearing that is used for many automobile and engineering parts is illustrated in Fig. 47. In this case the cone is provided with two grooves on its conical face, and the rollers are housed in a pressed metal cage.

LOW FRICTION SLIDING BEARINGS

It has always been a difficult problem, in engineering mechanisms, to provide the equivalent of a ball or roller bearing in the case of members having a purely sliding movement. The use of ball-bearing mounted rollers is one method of reducing sliding friction, whilst another is that of using a number of balls running in grooves between the fixed and sliding members.

Fig. 48 shows a special design of low friction bearing which is mounted in the fixed member by means of the cylindrical casing *E*, which is made a tight fit in a hole bored in the fixed member. The sliding member is in contact with the ball *A* which bears on a suitable groove *B* in the upper race of a thrust bearing having a series of balls *C* and a fixed race *D*. A cover *F* seals the bearing and serves to retain the lubricant. When the sliding member moves it rotates the ball *A*, which in turn rotates the upper thrust race and balls *C*, thus providing the minimum of frictional resistance.

For relatively heavy loads a roller *A* (Fig. 49) is employed in place of the ball. Owing, however, to the difference in linear speeds of the inner and outer radii of the roller *A*, the moving thrust race is made in two parts *B* and B_1, two separate series of balls *C* being used. The fixed races *D* and D_1 are also made as separate units for convenience of assembly.

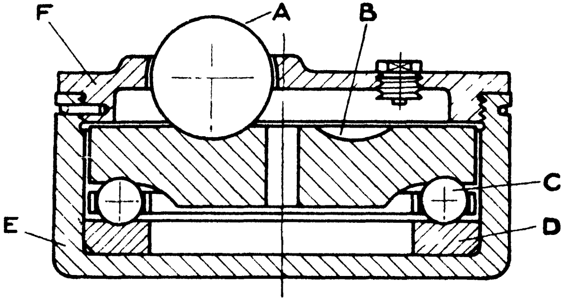


FIG. 48

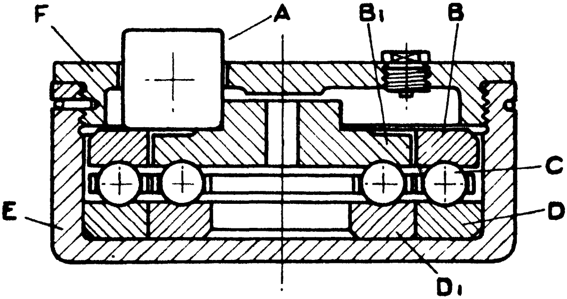


FIG. 49

THE HYATT FLEXIBLE ROLLER BEARING

Another design of roller bearing is the Hyatt type illustrated in Figs. 50 and 51. The principle of this bearing is quite different from that of any others we have described, for instead of employing solid ones in the bearing, the Hyatt type utilizes flexible rollers. These enable the bearings to accommodate themselves to small errors in alignment, and enable the rollers to bear evenly all along their tracks. It is claimed also that this type of bearing, on account of the resiliency of the rollers, will withstand impact shocks and load variation much better than solid roller bearings.

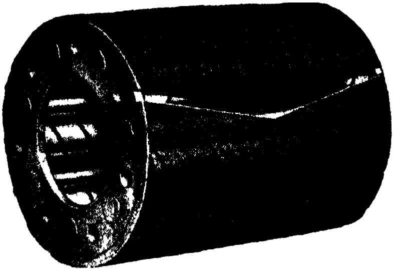


FIG. 50. THE HYATT FLEXIBLE ROLLER BEARING WITH SLIT OUTER RACE

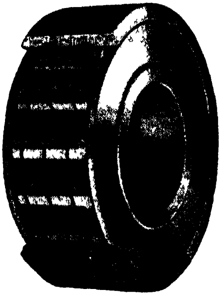


FIG. 51. THE NARROW TYPE HYATT BEARING

The rollers are in the form of hollow cylinders formed by helically winding strips of alloy steel. After being cut to length, the rollers are heat treated, not only for maximum hardness, but also to develop maximum toughness ; they are then very accurately ground to size. In this way very tough rollers offering maximum resistance to shocks, abrasion, and fatigue are obtained.

In assembly the consecutive rollers are arranged with right- and left-hand spirals (Fig. 51) ; this is for the purpose of directing the lubricant back and forth across the bearing surfaces, so as to ensure continuous lubrication of all parts.

The rollers are kept in proper alignment, and are regularly spaced by means of a cage or retainer. This consists of two pressed steel end rings held rigidly in parallel and concentric position by means of round steel spacing bars, the ends of which are riveted in countersunk holes in the end rings. The combination of rollers, cage, and bars is termed the *roller assembly* ; it is a unit that contains the rollers and can be handled separately, since the rollers cannot come out of place. The length of the roller assembly measured between outside faces of the end rings is always shorter than the outer race by a definite operating clearance, so that flat surfaces located directly against the ends of the outer race will not interfere with the operation of the bearing.

Two kinds of roller assembly are manufactured ; these are known as the " light " and " heavy " types.

In regard to the design of the *outer race*, there are two distinct types, known as the *Slit* and *Solid*, respectively.

The Slit Outer Race is shown in Fig. 50. It is used chiefly for light duties, the rollers being comparatively long ; it is designed primarily for direct roller operation. No inner sleeve or race is fitted, the rollers operating direct on the steel journal. A slit outer race is, however, provided ; this is of high carbon steel accurately finished to cylindrical form. The joint is in the form of an obtuse angle (Fig. 50), so that the rollers operate over it without shock or noise.

Before pushing into its housing the slit race will be found to have a slight outward spring, which causes the joint to open slightly. When brought to true cylindrical form by pushing into the machined cylindrical housing the joint is closed up to its proper width and the spring of the race eliminates any tendency to creep or turn in the bore. *The load-carrying capacity* of this type is governed to a large extent by *the hardness of the journal* upon which the rollers bear. This type of bearing has been widely used for back axle bearings of Ford cars.

The Solid Race Type Hyatt roller bearing (Fig. 51) is intended for heavy loads and higher speeds. In this case the rollers are fairly short, and both inner and outer solid races are provided.

As before, the spiral type of rollers with their cage is employed. The races are made in the form of straight cylinders of steel case-hardened all over to ample depth and ground to close limits.

The three components, namely, the inner and outer races and the roller assembly, are separable, and, if desired, either the inner or outer races can be dispensed with if suitable operating surfaces are substituted. This is a very useful feature in engineering work, for an appreciable amount of bearing space can be saved if one of the races can be dispensed with.

The correct size and type of bearing to employ in particular cases will depend upon the speed, shaft hardness (where no inner race is used), nature of service, and the maximum temperature to which the bearing is exposed. Hyatt bearings will operate on the normal rating basis up to 350° F. Above this temperature the temper of the races is affected, and the load capacity therefore reduced. Thus at 600° F. the capacity is reduced to one-third.

The manufacturers supply tabular and graphical data in regard to the appropriate sizes of roller bearings to work under specified conditions.

The slit race bearing is obtainable in a range of sizes varying from $\frac{1}{2}$ in. diameter shafts (with corresponding working loads of 62 lb. at 100 rev. per min., and 33 lb.

at 1000 rev. per min) up to 5 in. diameter shafts (with corresponding working loads of 5400 lb. at 100 rev per min. and about 3000 lb. at 500 rev per min. for bearings of 6 in. width).

HYATT BEARING MOUNTING NOTES

The principal points to observe when designing mountings and housings for Hyatt bearings can be enumerated, briefly, as follows—

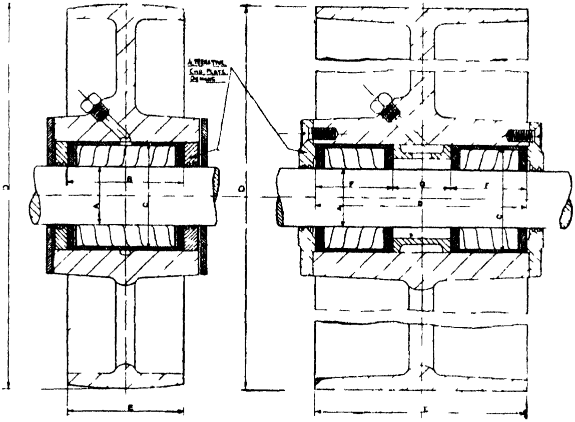


FIG. 52. SHOWING APPLICATION OF HYATT BEARINGS TO PULLEYS AND SHAFTS

1. *Correct alignment of bearings and shafts.*

2. *Retainment of the bearing parts* in their correct operating position by means of suitable collars, shoulders, sleeves, or recesses, as in the case of ball bearings.

3. *Endwise location of the shaft.* Hyatt roller bearings do not take end loads, so that it is necessary to provide some means to locate the shaft in its correct lengthwise position and to take any end thrust that may occur.

For small end thrusts and at low to normal rotational speeds, plain metal thrust washers may be used. For appreciable end thrusts, ball thrust washers are necessary. Sometimes a deep groove location journal ball bearing can with advantage be used.

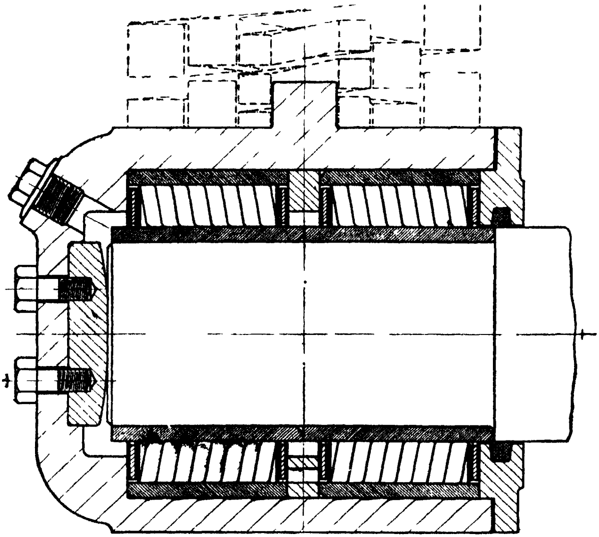


FIG. 53. HYATT BEARINGS AS USED FOR RAILWAY CARRIAGE AXLE BEARINGS

4. *Protection of the bearing.* As in the case of other ball and roller bearings, some means must be provided to exclude dust and water and to retain the lubricant in the bearing. The most common method for a blind end shaft is that of an end cap or plate sealing this end of the bearing.

For bearings having shafts passing through, a cover ring having a small clearance around the rotating shaft, and provided with a series of small grease-retaining grooves, is the most commonly used. Sometimes, as in the example illustrated in Fig. 53, a felt retaining washer is housed in the cover ring. The illustration referred to shows the arrangement of a railway carriage axle running on two sets of Hyatt bearings, in place of the usual plain gunmetal axle boxes ; the friction is thus considerably reduced. It should be noted that there is a plain metal disc to take the side thrust, and that the "open" end of the bearing has an inserted felt washer to retain the lubricant and to exclude dirt and water. The complete bearing is packed with grease, injected or replenished through the hole shown with its screwed plug on the left.

5. *Lubrication*. As with ball bearings the lubricant is intended, not to reduce friction of the rollers in their races, but to protect the surfaces against corrosion, and also to act as a seal to exclude dirt and moisture. The lubricant does, of course, tend to reduce friction of the cage and rollers.

As a general procedure, and in cases where the lubricant can be properly replenished, *oil* should receive first consideration ; after this a thin grease, and finally a thick grease.

If the temperature of the bearing is high a suitable high viscosity lubricant should be employed ; usually a cylinder oil is used for temperatures of 400° to 600° F. A cup grease is recommended for temperatures between 32° and 180° F. The use of greases containing any solid fillers such as cork, asbestos, graphite, or mica should be avoided, as the solid material is rolled into flakes and is harmful to the bearing.

NEEDLE BEARINGS

A disadvantage of the ball and roller types of journal bearing is that of their somewhat excessive overall diameter as compared with the plain bush pattern bearing. For this reason it is not always possible to substitute such bearings for the ordinary type. The increased diameter necessitates a corresponding greater size of housing, and this in turn results in greater bulk and weight of the component concerned. In order to overcome this difficulty in instances where the beneficial properties of reduced friction combined with lightness and small dimensions are concerned, a bearing of the needle pattern is now employed (Fig. 54). A typical instance of the application of the latter design of bearing is for the small end bearings of a petrol engine connecting rod—where an ordinary journal ball bearing could not be used on account of its large dimensions.

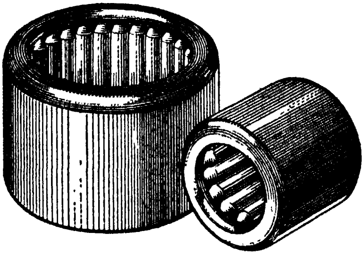


FIG. 54. TYPICAL NEEDLE BEARINGS

The needle bearing is a kind of roller bearing having very small diameter rollers or "needles" which, however, are not separated by cages or retainers, but are inserted direct between the inner and outer races. The small diameter rollers do not, as might at first be thought, rotate at very high speeds, but they act as a kind of floating sleeve experiencing only a limited amount of rotation under changes of bearing loads. The cuneate or wedge-shaped spaces between the needles induce capillary action, thus maintaining the oil in constant circulation and preserving the oil film which is essential to their satisfactory operation.

The needles are usually made of a chrome steel, heat treated so as to give a Brinell hardness of 600 to 700. Owing to the relatively greater number and length as compared with ordinary roller bearings they give a much larger contact area than the latter, and so are able to withstand higher loads per unit area; in this connection bearing loadings of 4500 lb. per sq. in. of (projected) bearing area are permissible with needle bearings. In their application, whilst a little radial clearance is permissible—when operating conditions allow—it need not be restricted quite so much as with

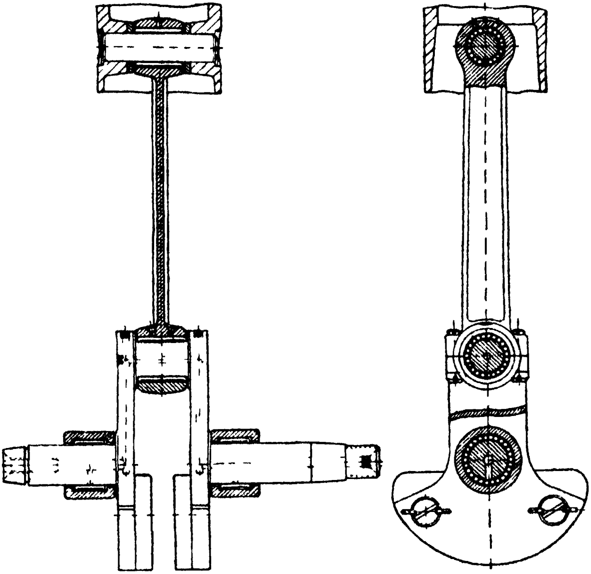


FIG. 55 CONNECTING ROD NEEDLE BEARING

roller bearings, and may vary from 0·001 to 0·003 in. according to the size of bearing. It is important to ensure that the needles are not fitted in such a way as to be tight in the housings, nor must they fully occupy the annular space between the latter. In filling this space with needles it is immaterial whether there is insufficient space to insert the last needle so that it is not necessary to proportion the diameters of the inner and outer housings to take an exact number of needles.

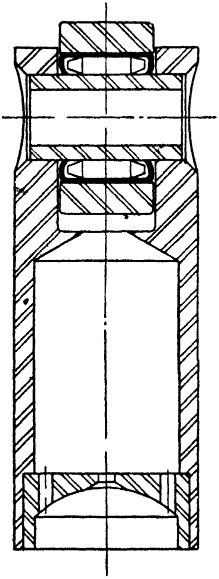


FIG. 56
VALVE TAPPET

The matter of adequate lubrication is, however, of importance since the successful operation of this type of bearing depends upon the maintenance of the oil film all the time. In this connection the use of oil—not grease—is recommended, but the surfaces of the needles and races may with advantage be given a light coating of grease to facilitate assembly.

The standard needles in the bearings range from 2 mm. to 4 mm. as a rule, and the length is usually made from 6 to 10 times the diameter. Typical applications of needle bearings are shown in Figs. 55 and 56. The former illustration shows the application of these to a petrol engine connecting rod, and the latter to the roller at the lower end of a valve tappet. Owing to its high load capacity and impact resisting qualities this type of bearing has proved quite satisfactory under the most exacting conditions for such purposes, as well as for the bearings of rocker arms of valve gears, for gear-box main-shaft bearings and similar applications.

Usually the surfaces of the outer and inner races are of case-hardened alloy steel, but in certain instances—as for shaft journals, when the latter are surface-hardened—the inner race can be dispensed with so that the needles run directly on the journal itself.

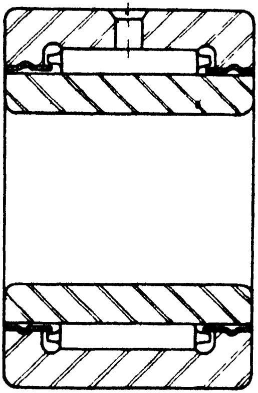


FIG. 57

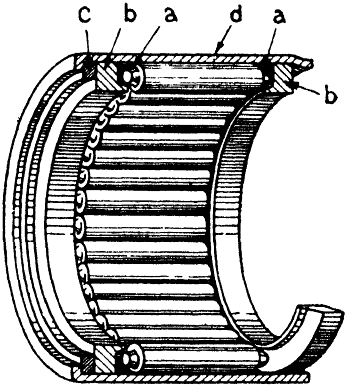


FIG. 58

Fig. 57 shows the Ransome and Marles improved design of needle bearing which utilizes needles having trunnion ends; these enable more simple needle retainers to be employed. The bearing shown consists of the usual outer race with shoulders for guiding the rollers. Pressings are fitted in the bore of these shoulders and the trunnions of the rollers resting on these pressings are retained in the outer ring. Fig. 58* shows another design of assembly for use where the rollers are required to run between suitably hardened and ground surfaces of a machine without the use of separate race rings. The rollers *a* are held between suitably formed end plates *b* which prevent them from falling towards the centre; a spring ring *c* retains the assembly in position after the bearing is fitted. The whole is held by means of a thin metal container *d* which is peened over at the end nearest to the spring ring, whilst three or more notches are formed in the opposite end to retain the assembly in the cartridge.

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* *The Engineer.*

THE LUBRICATION OF BEARINGS

In the cases of both ball and roller bearings, as the action of the balls or rollers is purely one of rotation and not of sliding, the question of lubricating the balls or rollers is unnecessary as a means of reducing friction. It is clearly not possible to maintain a film of lubricant between the ball or roller and its races, owing to the relatively great intensity of pressure.

The principal object, however, in lubricating these bearings is to protect the highly polished surfaces from corrosion and to keep out dust and water. Either grease or oil will do for this purpose, provided that it can be kept in the bearing. In the majority of cases it is found more convenient to use grease, as this will remain in the bearing for long periods without needing replenishing. The cages are also lubricated by the oil or grease supplied, to reduce sliding friction. A lubricant the consistency of vaseline, *non-acid* in character, is suitable for most conditions and temperatures up to 120° F. The melting point should not be lower than about 200° F.

In cases where the bearing is exposed to higher temperatures either a high melting point grease or a mineral lubricating oil should be used.

Animal and vegetable greases or oils are apt to develop free acid, and therefore should be avoided in ball and roller bearings.

In many cases oil is preferable to grease, notably where bearings run at high speeds, and may be subjected to higher working temperatures. Petrol engine bearings are a case in point. When ball bearings are used in gear-boxes and certain engine bearings, they are lubricated with oil in preference to grease.

PROTECTION OF THE BEARINGS

Apart from the use of a grease lubricant to keep out water and small dust particles, it is usually necessary to provide additional means for protecting ball and roller bearings from ingress of dust and water, or these agents would quickly cause abrasion and corrosion. Moreover, the lubricant must be kept within the bearing also. It is usual to make provision in the design of the housing or shaft for this protection. One common method is to provide a wide lip to the bore of the end cover and to make it a fairly close fit to the shaft, with a clearance of about $\frac{1}{100}$ in. In the bore a series of grooves are turned (Fig. 59), which become filled with grease and act as a seal, effectively excluding dirt and water. Semicircular grooves of about $\frac{1}{16}$ in. diameter are found to give good results.

For particularly dusty conditions the addition of a thrower over the lip of the end cover (Fig. 60) is required.

Fig. 60 shows an excellent form of bearing protector, which has proved most satisfactory under conditions similar to those of axles on motor vehicles in excluding grit and water. This combination of a thrower and two grooves is both simple and effective under severe conditions of bearing operation.

When bearings have to be protected against the force or pressure of water, they should be packed with grease, and a grease gun used for forcing the grease into place. Many motor-car parts are thus protected. Fig. 61

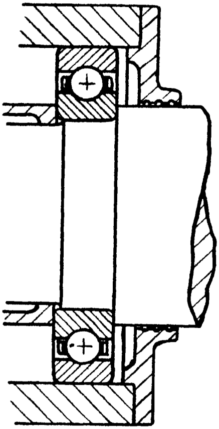

FIG. 59

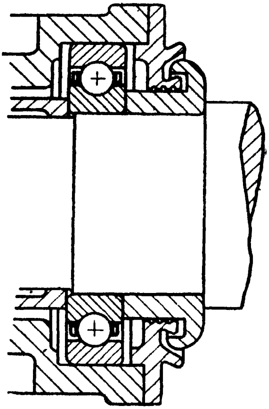

FIG. 60

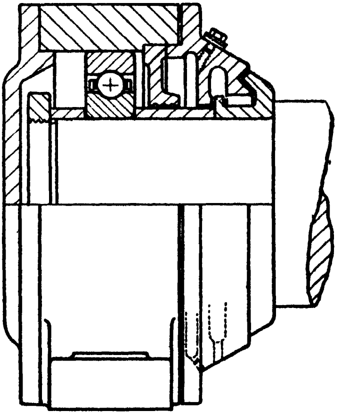

FIG. 61

ILLUSTRATING ALTERNATIVE METHODS OF PROTECTING BEARING ASSEMBLIES illustrates a good design of bearing (Hoffmann) protection for such cases.

The worst conditions likely to be met with are those of bearings running totally immersed in water or other liquids. In such cases, apart from the adoption of the best available design of housing protection, stainless steel bearings are advisable. Although more costly and of lower load capacity, the non-corrosive property of this steel renders it the most suitable. Fig. 62 illustrates some alternative protective measures or *closures* recommended for Timken roller bearings; these employ respectively (from left to right) felt washers, sheet metal labyrinth discs, and annular grooves.

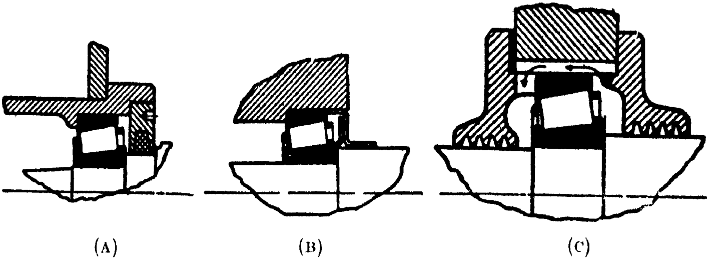
(A) (B) (C)

FIG. 62. ILLUSTRATING THREE DIFFERENT METHODS OF PROTECTING ROLLER (AND BALL) BEARINGS

A = Felt disc *B* — Sheet metal discs *C* = Annular grooves

Other examples of bearing protection will be found among the illustrations of this section.

SECTION XXXI

PETROL ENGINE FITTING AND ASSEMBLY

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SECTION XXXI

PETROL ENGINE FITTING AND ASSEMBLY

INTRODUCTION

THE two main types of petrol engine are the stationary and the transport. The stationary engine is required to run at a more or less constant speed for long periods, and the design and operation are very similar to those of a gas engine. The transport engine, however, works under conditions which add considerably to the problems of design and maintenance. First there is the reduction in weight to be considered, and secondly the speed of normal working is variable, being anything between 500 r.p.m. and 4,500 r.p.m. or more for small engines. At any of these speeds the actual load may be small or as big as the engine can carry, and, under all these conditions, the efficiency must be reasonable. Aero engines are built for lightness, but may be considered constant-speed engines, the speed being round about 2,200 r.p.m. Marine engines, except those in use in very small craft, may be classed with the stationary type of engine.

This section will deal almost entirely with the transport engine as the fitting of the stationary type is dealt with in the section on gas engines (Section XXXII).

The style of fitting required is very different from that on larger engines; there is little filing and bench work, but a great deal of careful and intelligent adjustment is needed. Replacement parts usually are easily obtained, and will work after a fashion if just "put together." But to make a good job, these parts, although standardized and produced by clever machine processes, need accurate and skilful fitting.

USE OF TOOLS

Firstly, a good mechanic can always be told from an inspection of his tools. The tools need not be numerous, although the out-of-the-way tool may be invaluable once in a while. The tools are clean, and do not show signs of being mis-used: mis-use of a tool is not a crime if, after due consideration, it is expedient in the circumstances.

Spanners which do not fit the nut ruin the nut and the spanner too: worse, a badly fitting spanner may slip off the nut and cause the user to bark his knuckles. This is bad enough, but it also destroys that tranquillity of mind which is very essential for sound and methodical work.

In tightening a nut, it is not necessary to overstrain it. The leverage of a spanner is about six to eight times the distance across the jaws, and except in special cases, normal manual pressure without extra leverage will suffice to tighten the nut. Experience only can develop the "feel" of a nut. If the nut must be turned a little farther in order to insert a split pin, then the base of the nut should be filed a little, or a thinner washer used, or even the nut exchanged for another, rather than force the nut to the required position. Of course, to slacken back the nut to the last hole to get the split pin in is criminal practice. Box spanners must also be the correct size when used.

Hammers are very useful implements in the right hands. Unfortunately the use of these tools is greatly misunderstood, because it needs experience to employ a hammer correctly. A light hammer with a "bouncing" tap will rivet the end of a pin without driving the pin. On the other hand, a heavier hammer with a "follow-through" stroke will drive the pin without causing riveting. Discretion must be used, but only experience can dictate the weight of the blow required. Bronze sticks, copper or hide mallets or hard wood may be needed to prevent bruising.

A tight nut may have to be chopped off as a last resort, but a good soak with paraffin or penetrating oil should be tried first. If the nut is accessible, a smart blow struck squarely on the face (not the top) of the nut will usually free it.

Probably one of the most frequently used tools is the screwdriver. The blade of this should always be fitted into the slot of the screw properly before attempting to turn the screw. If the screw is reluctant to move, it is advisable to strike a blow with a hammer on the end of the screwdriver when in position. The screwdriver must of course be of the engineer's type with the blade passing through the handle. Extra leverage may be applied if necessary, but care must be taken to see that the screwdriver really fits the slot first. Nothing looks worse (or is a greater nuisance) than a torn slot in a screw head.

Cleanliness may sound fussy, but the time taken to brush a part of an engine down with paraffin and wipe clean is negligible, and is amply repaid. Orderliness is very important as parts easily get mislaid. Small parts as soon as removed should be dropped into suitable trays containing "washing" paraffin. Generally it is not necessary to keep each nut and bolt together, but more will be said on this point under bearings.

Further, the order in which an engine is dismantled is of importance, and if the maker's instructions are available they should always be well perused before starting the job—not in the middle of it. On a strange engine it may be advisable to make a note of the order in which certain parts have been dismantled to facilitate assembly.

Since wear and mis-use of parts give much more information than the assembly of the new engine, experience in a first-class repair depot is to be preferred to that obtained in a construction factory.

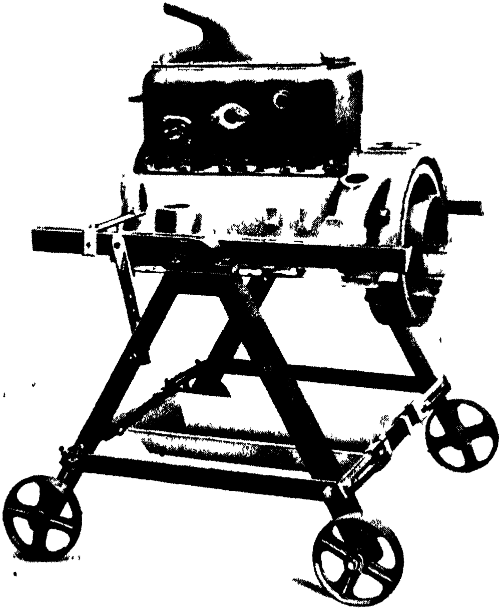
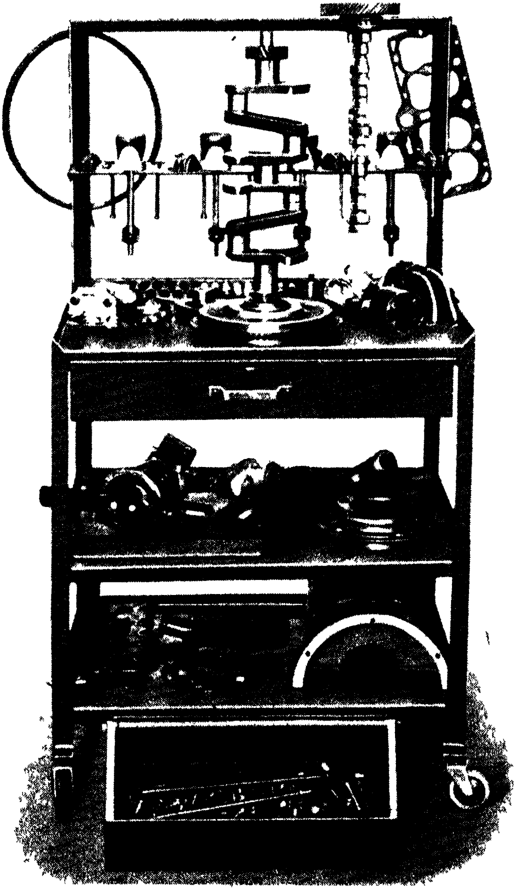


FIG. 1. REVERSIBLE STAND ADJUSTABLE FOR ENGINES OF ALL SIZES

A good example of a stand adaptable for engines of all types or sizes is made by Mann, Egerton & Co., Ltd. (Fig. 1). The engine may be bolted to side members carried by trunnions on the triangular side frames, the



(Courtesy of the Ford Motor Co Ltd)

FIG 2 PARTS CARRIER

side members having undercut grooves for the securing bolts. The engine is held in any desired position, upright or inverted.

The parts carrier shown in Fig. 2 loaded with the parts of an engine is very useful, but method must be adopted or the carrier soon becomes a portable rubbish dump. Other types of carrier are arranged with a vice in position.

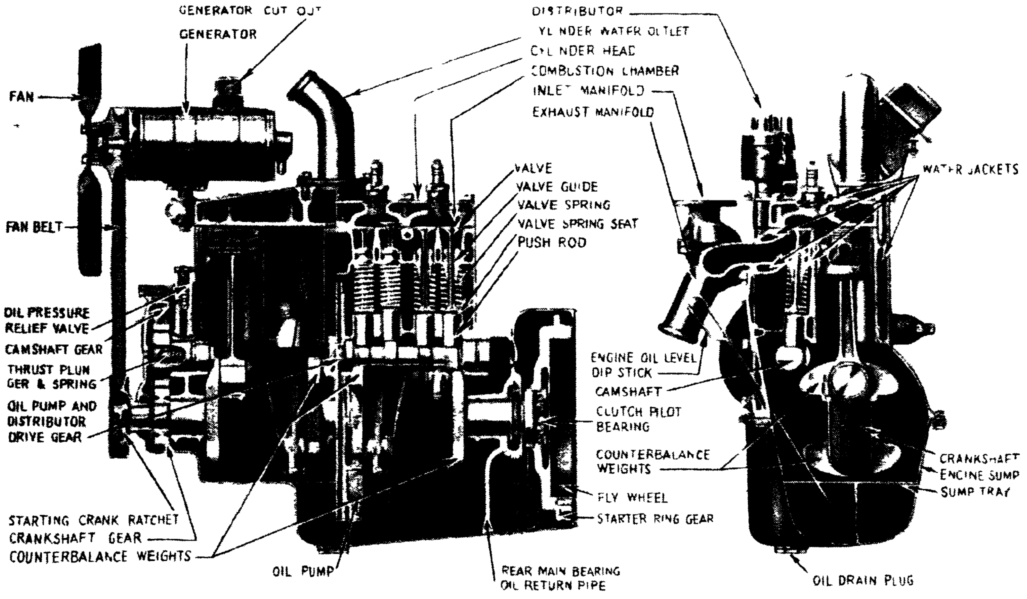
TYPES OF ENGINE

It is quite impossible in the scope of this section to deal with every engine manufactured, and only typical examples will be noted. The Ford 8 h.p. engine is shown in the two sectional views in Fig. 3.

Notice the heavy main bearings to the crankshaft and the balance weights; the valves are inclined to the vertical axis of the engine. The sectional views show the construction of the detachable head clearly. The drive for the oil pump and distributor is taken from the camshaft by skewgears. The clutch and gear-box are shown disconnected, but when the gear-box is bolted to the clutch housing, the whole forms a complete unit which is carried on three points in the chassis by a cross-bearer in the front of the engine and by a support at the gear-box.

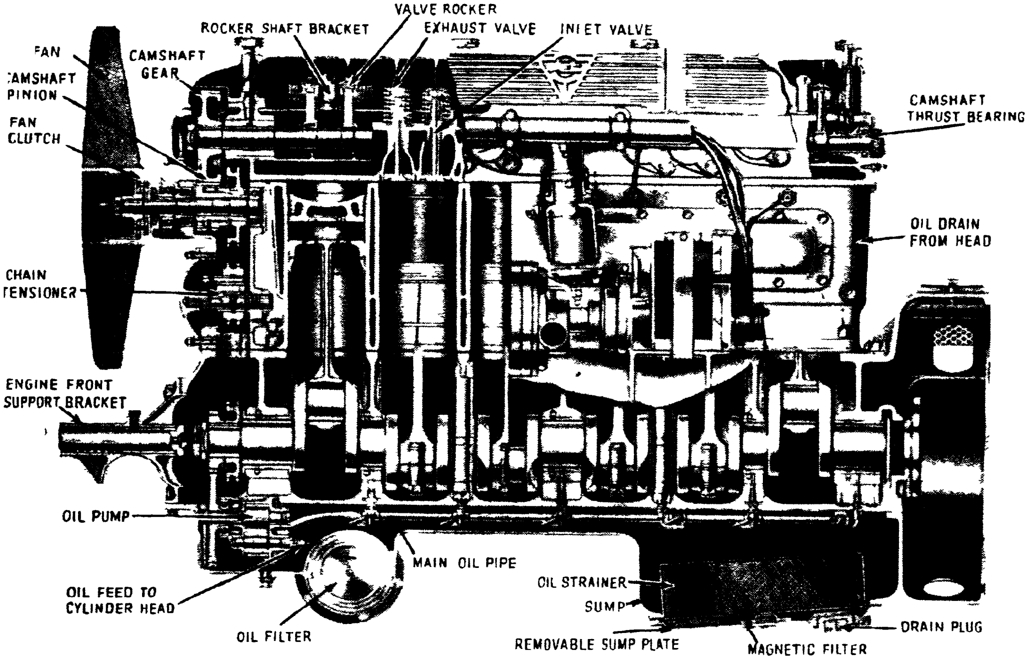
A much larger engine is shown in Fig. 4. This is a 120 h.p. six-cylinder engine made by the Associated Equipment Co., Ltd.

Many engines of this type have a maximum speed determined by a governor; and in this case the speed control is at 2,400 r.p.m. The balance of a six-cylinder engine is better than that of four cylinders and generally balance-weights are not fitted as in this case. Here the valves are overhead and the camshaft operates the valves through bell-cranked levers as shown in Fig. 5.



(*Courtesy of the Ford Motor Co Ltd*)

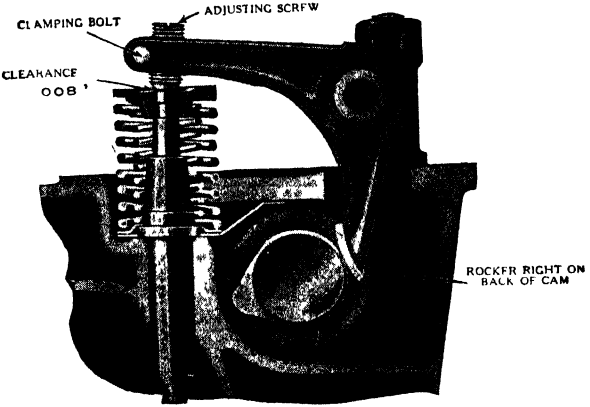
FIG. 3. SECTIONAL ELEVATION AND END VIEW OF FORD 8 H P ENGINE



(*Courtesy of A E C, Ltd*)

FIG. 4

Notice the magnetic filter for removing particles of steel, and also the oil filter in the main oil supply. The lubrication system can also be seen to each main bearing, thence to each crank pin. The construction of the piston with the supporting struts and the split skirt should be noted.



(*Courtesy of A E C , Ltd*)

FIG. 5

A stationary single-cylinder engine made by the National Gas Engine Co., Ltd., is shown in Fig. 6.

The generous design of all parts to give long and reliable service should be contrasted with the quite different aspect of the design of the previous engines. All are first-class designs to meet particular requirements.

As a last example, the Douglas two-cylinder motor cycle air-cooled engine is shown in part section in Fig. 7.

The inlet valves are at the top and the exhaust valves at the bottom. The pistons are opposed, that is, the crank pins are at 180°, so that both pistons go outwards together. This gives excellent balance.

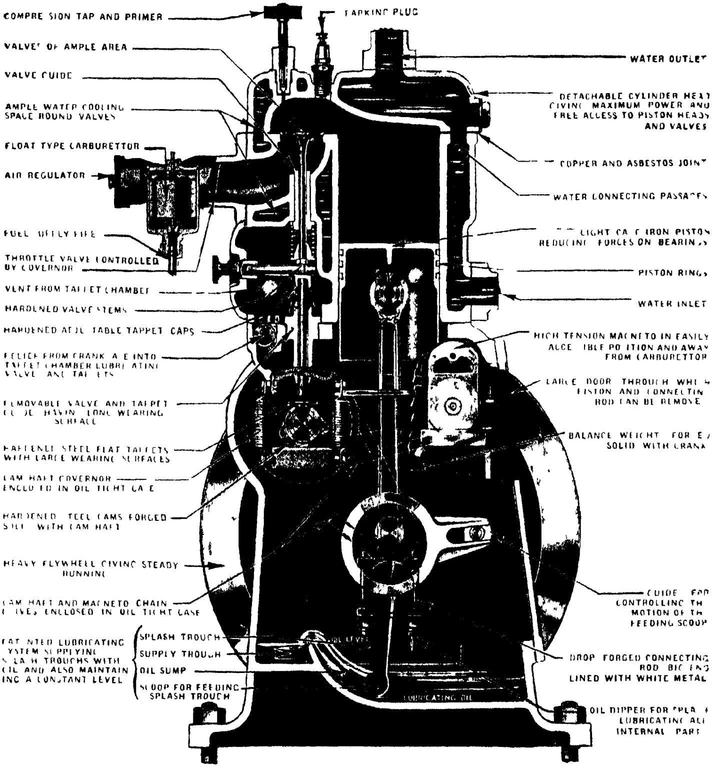

COMPRESSION TAP AND PRIMER
SPARKING PLUG
VALVES OF AMPLE AREA
VALVE GUIDE
AMPLE WATER COOLING SPACE ROUND VALVES
FLOAT TYPE CARBURETTOR
AIR REGULATOR
FUEL SUPPLY PIPE
THROTTLE VALVE CONTROLLED BY GOVERNOR
VENT FROM TAPPET CHAMBER
HARDENED VALVE STEMS
HARDENED ADJUSTABLE TAPPET CAPS
RELIEF FROM CRANK CASE INTO TAPPET CHAMBER LUBRICATING VALVES AND TAPPETS
REMOVABLE VALVE AND TAPPET GUIDE HAVING LONG WEARING SURFACE
HARDENED STEEL FLAT TAPPETS WITH LARGE WEARING SURFACES
CAM SHAFT GOVERNOR ENCLOSED IN OIL TIGHT CASE
HARDENED STEEL CAMS FORGED SOLID WITH CAM SHAFT
HEAVY FLYWHEEL GIVING STEADY RUNNING
CAM SHAFT AND MAGNETO CHAIN DRIVES ENCLOSED IN OIL TIGHT CASE
PATENTED LUBRICATING SYSTEM SUPPLYING SPLASH TROUGHS WITH OIL AND ALSO MAINTAINING A CONSTANT LEVEL
SPLASH TROUGH
SUPPLY TROUGH
OIL SUMP
SCOOP FOR FEEDING SPLASH TROUGH
OIL LEVEL
LUBRICATING OIL
WATER OUTLET
DETACHABLE CYLINDER HEAD GIVING MAXIMUM POWER AND FREE ACCESS TO PISTON HEADS AND VALVES
COPPER AND ASBESTOS JOINT
WATER CONNECTING PASSAGES
LIGHT CAST IRON PISTON REDUCING FORCES ON BEARINGS
PISTON RINGS
WATER INLET
HIGH TENSION MAGNETO IN EASILY ACCESSIBLE POSITION AND AWAY FROM CARBURETTOR
LARGE DOOR THROUGH WHICH PISTON AND CONNECTING ROD CAN BE REMOVED
BALANCE WEIGHT FOR E
SOLID WITH CRANK
GUIDE FOR CONTROLLING TH
MOTION OF TH
FEEDING SCOOP
DROP FORGED CONNECTING ROD BIG END LINED WITH WHITE METAL
OIL DIPPER FOR SPLA
LUBRICATING ALL INTERNAL PART

FIG. 6. NATIONAL SINGLE-CYLINDER STATIONARY ENGINE

DISMANTLING THE TRANSPORT ENGINE

If possible the oil should be drained from the crankcase immediately after the engine has been running for some time. Remove the plug so that the sump can flush itself clean quickly rather than allow the oil to drip slowly. It is a good plan to allow the sump to drain overnight, and then remove the tray full of dirty oil first thing in the morning. The radiator should be drained.

Whilst the engine is still hot it is a good plan to attempt to loosen the exhaust flange and manifold nuts and the cylinder head nuts as they may be easier than when cold.

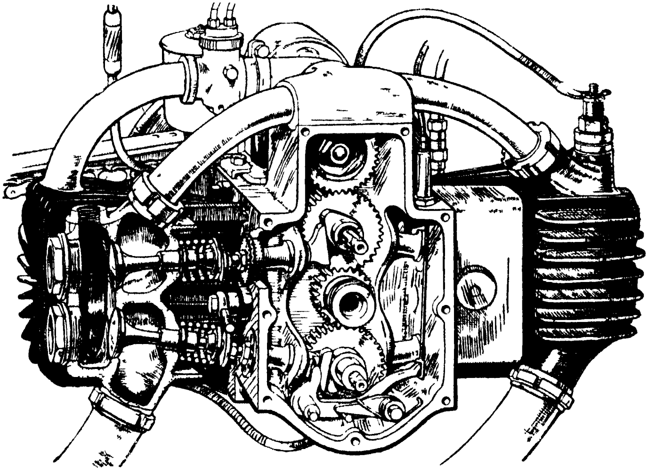


FIG. 7. DOUGLAS TWO-CYLINDER ENGINE

In the morning the bonnet should be removed and the wings of the vehicle covered to prevent marking. If the engine is to be removed from the chassis, the radiator will need to be removed also.

The order in which the various connections to the carburettor, ignition, dynamo, etc., will need to be removed must be left to discretion, but before the electrical side is tackled one of the main battery terminals should be disconnected to prevent accidents. Make sure that petrol does not siphon from the tank when the petrol pipe union to the petrol pump is undone.

When the engine is to be removed completely, in many cases the dynamo, carburettor, manifold, etc., may be left in position. The engine should be supported by suitable lifting tackle which "just takes the weight" before the clutch housing is unbolted, and when the engine is ready to be drawn forwards from the gear-box, care must be taken to see that no clutch fitments fall out from the clutch housing when this is done.

If the engine is to be left in the chassis it is as well to remove any components which may be damaged or may be in the way when working. Generally the instruction book will be available, and this should be followed carefully.

The manifold nuts if not slackened as previously explained may present difficulties. They are usually brass nuts and may require a little care. Sometimes it is not necessary to remove the carburettor first, but if there is any doubt, this should be done, as the carburettor will need to be dismantled later for cleaning. Do not drive a screwdriver between the manifold and cylinder so as to destroy the gaskets, or try to get one side of the manifold off first. Often a lip is provided for wedging; if not, choose the places for wedges carefully so as not to spoil machined surfaces. The gaskets may often be used again.

If in removing one of the nuts the stud should be drawn instead, this does not matter; it may be put back on assembly in the same way, but it is better to free the nut on the bench and fit the stud properly. A stud which "bottoms" and leaves some of the thread above the top of the casting is fitted wrongly.

The last thread of the stud should jam in the top of the hole.

Plug leads should be clearly marked with the number of the cylinder to which they belong, if this is not already done.

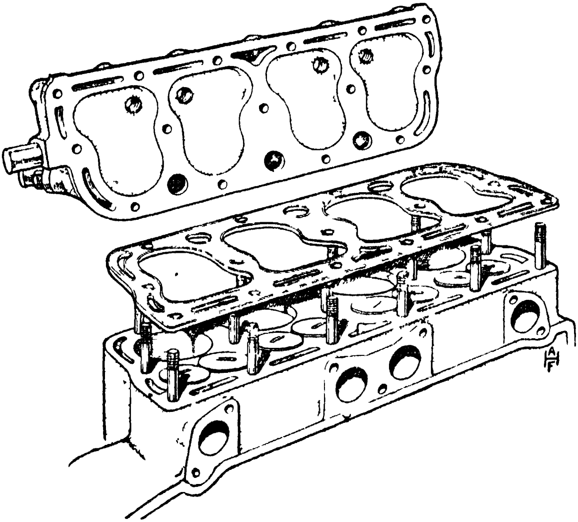


FIG. 8. DETACHABLE CYLINDER HEAD AND GASKET

THE CYLINDER HEAD

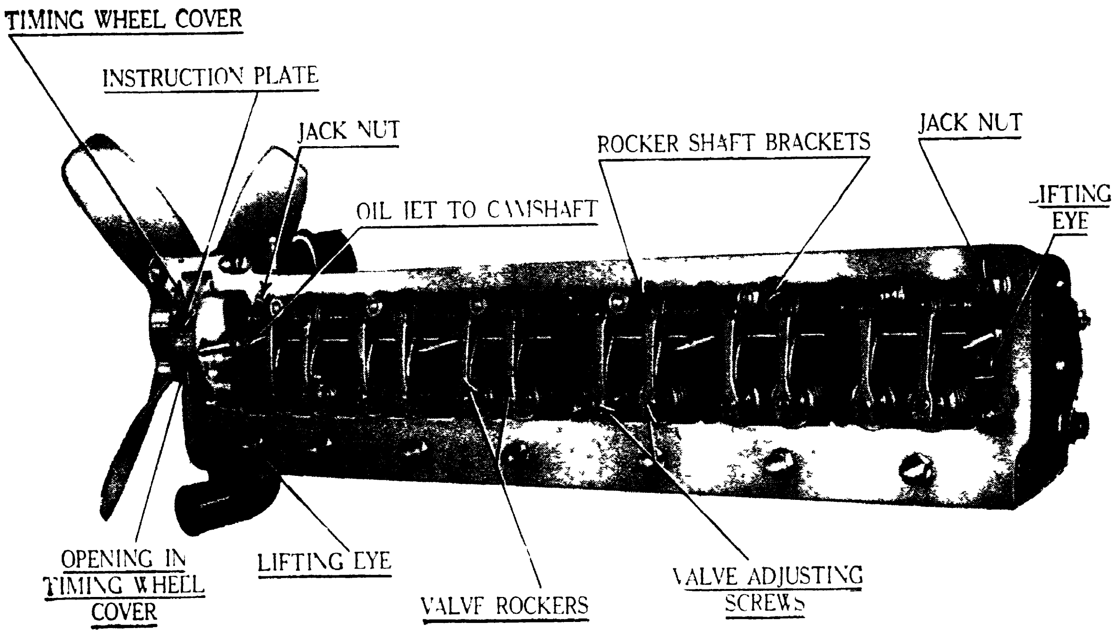
The removal of the cylinder head may present difficulties. With the side-valve engine there is no need to consider the timing of the valves, but in the type of engine shown in Fig. 4 with overhead camshaft, this must be considered, although it is possible to re-time the valves as shown later. In this engine the flywheel can be locked when the pistons are on dead centre by means of a special spring bolt. In other cases the markings on the flywheel and the markings on the camshaft drive must be observed. If necessary, turn the engine over slowly once or twice to familiarize yourself with the timing and the timing-marks. It is never correct to "centre-pop" your own markings.

If the cylinder head is fitted with jack-nuts as in Fig. 9 the cylinder head nuts may all be taken off and the joint broken by using the jack-nuts. Make sure first, however, that *all* the cylinder head nuts are off; there may be one or two which carry camshaft bearings which may have been overlooked. The lifting eyes may be used for the lifting tackle, but a long sling should be used. Gently rock and ease the head whilst lifting. If there are no jack-nuts, it is a good plan to use the compression of the engine to lift the head. This is often done whilst the engine is left in the chassis. Judicious use of a mallet will usually help if the above method fails. The gasket should be freed carefully with a rule or a knife as lifting proceeds, and usually it will be kept in a good condition for re-use. Do not bruise the machined surfaces with a chisel. In extreme cases it may be necessary to use wedges, but great caution must be exercised.

When the head is removed, examine the gasket carefully and see whether there has been any blow-by of gases, or whether there has been water leakage. Also see that all water passages have been clear so that this can be rectified on assembly.

The cylinder head is a complicated casting and should be examined to see whether it has warped. In the latter case it may be desirable to have the surface machined a small amount.

The order of the further dismantling of the engine must be left to the Instruction Manual, and the general care and treatment of the several items will be considered. In some engines the pistons and connecting rods can be removed through the crank-case, for



(Courtesy of A E C Ltd

FIG. 9. VIEW OF CYLINDER HEAD

example, and every engine must be considered according to the design.

PISTONS

Most pistons are to-day made of aluminium alloy, which besides being lighter than cast iron is a better heat conductor. This latter point is of utmost importance, as with good heat conduction the distortion and internal strain are reduced. Aluminium has a large coefficient of expansion, but when alloyed suitably the expansion can be made to approximate to that of cast iron. The surface of most alloy pistons is anodized to give a very hard surface and also to help the lubrication by adsorbing the oil. Cast-iron pistons in cast-iron cylinders obtain a glasslike surface with wear and proper treatment; alloy pistons, however, give very good results when carefully designed and fitted.

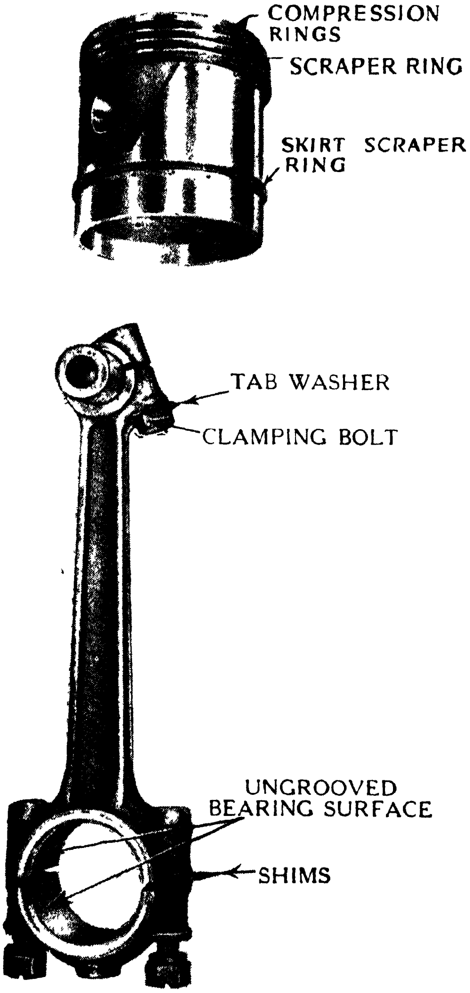
The piston in Fig. 4 is also shown with its connecting rod in Fig. 10. There are three compression rings and two scraper rings, one above the gudgeon pin and one on the skirt. The piston is split to allow for expansion, and the split must *always* go towards the near-side of the engine, so that the plain side is the one which carries the explosion thrust.

The amount of clearance between the piston and the cylinder varies with the material of the piston and the type of engine. At the skirt and over the parallel portion of the piston the clearance is about 0·001 in. per in. diameter, but above this the clearance is greater owing to the higher temperatures. This is shown diagrammatically in Fig. 11.

The upper part of the piston is usually about 0·0035 in. per in. of diameter smaller than the cylinder diameter.

THE PISTON RINGS

The compression rings (usually two or three in transport engines) are of cast iron, and in most modern



(*Courtesy of A.E.C., Ltd*)

FIG. 10. CONNECTING ROD AND PISTON

engines the top ring has a gap when tried in the cylinder of about 0·002 in. per in. diameter. The joint may be square or "scarfed," and the clearance should not be too small at the gap as this will cause the ends to butt before it gets properly heated and so fracture the ring. The top ring, which gets the hottest, should have a clearance in the groove of about 0·003 in., whereas the others, including scrapers, should have a clearance of about 0·002 in. If this clearance in the groove is not provided, the ring will wedge itself in the groove and will not function. Fig. 12 shows the method of trying the rings in the grooves, and Fig. 13 the method of checking the gaps which are here shown as "scarfed."

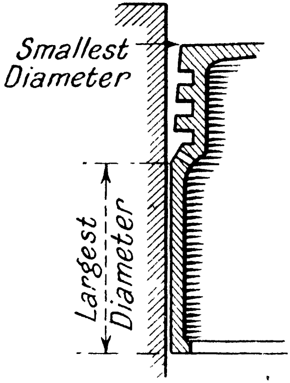


FIG. 11. PISTON CLEARANCE

The scraper rings most commonly used are shown in Fig. 14 *B*. The type *A* is sometimes adopted. The main trouble with scraper rings is that they wear rapidly as the surface is small, but the "springiness" is almost the same as that of the compression ring. The holes in the ring and the drain holes in the piston tend to become clogged with use. These drain holes can be clearly seen in Fig. 10.

REMOVING AND FITTING THE RINGS

Very often it is possible to use the rings again, but they must be taken off the piston to be cleaned. They must be kept in order, and as it is not feasible to mark them, special care must be taken to see that they do not become confused. Sometimes a new top ring only is fitted, as this is the one which receives most wear under the most trying conditions.

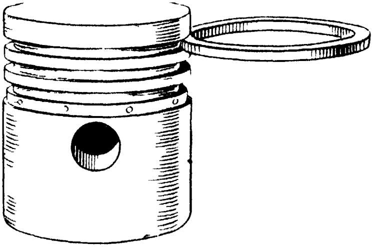


FIG. 12 TRYING PISTON RINGS

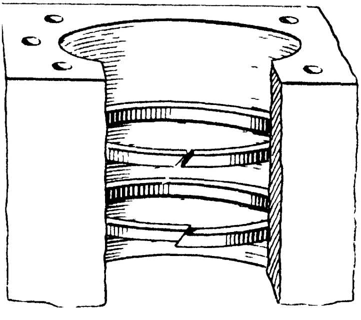


FIG. 13 FITTING PISTON RINGS

The skids used to remove the rings as indicated in

Fig. 15 may conveniently be made of pieces of hacksaw blades with the teeth ground off and the edges rounded. Start at the gap and work one skid round to the back,

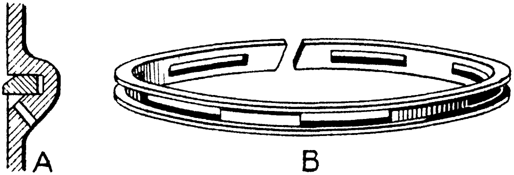


FIG 14. SCRAPER RINGS

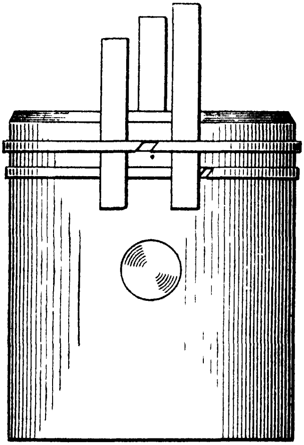


FIG. 15. RING REMOVAL

then insert the other two (or three) and the ring can be worked off the piston. Although cast iron is very brittle, it will take a permanent set if stressed nearly to breaking point, and rough handling will spoil the circularity of the ring.

The reverse procedure is adopted for replacing the rings; but first the gap must be tested carefully with the ring in the cylinder and the side clearance of the ring tested in the groove, as in Fig. 12.

If when trying a new ring it is too tight in the groove it may be rubbed down, as in Fig. 16. Great care must be taken to keep the ring flat and uniform in thickness.

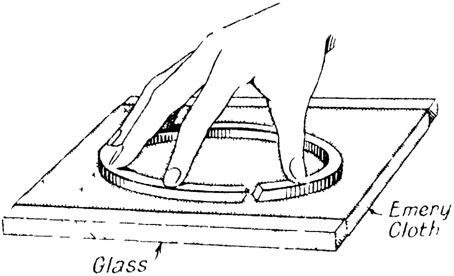


FIG. 16. RUBBING ON EMERY CLOTH

When the piston is being replaced in the cylinder, it is generally necessary to use a compressor as in Fig. 17. A more elaborate compressor is shown in Fig. 18.

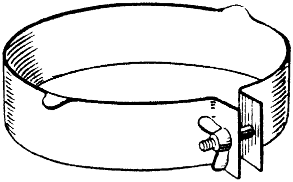


FIG. 17. PISTON RING COMPRESSOR

The bottom of the cylinder has a counterbore which facilitates entry of the rings. If, however, the piston must be put in from the top of the cylinder, a compressor is necessary as in Fig. 19. It is easy to break a ring, and the utmost care must be taken; ease the piston and ring continually during entry, and keep the fingers clear as the rings can cause a nasty cut. If there is any doubt about the satisfactory entry of the rings, the piston must be withdrawn again; do not "hope for the best." The gaps in the ring should be equally spaced around the circumference; but as the gaps will always try to work together in use, some makers dowel the rings. This adds greatly to the care required in fitting.

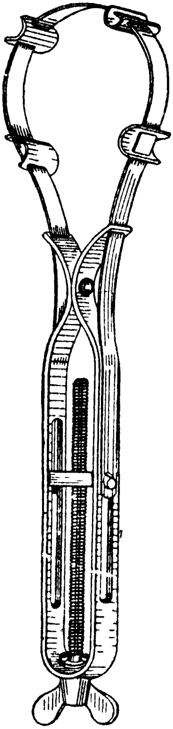


FIG. 18. ANOTHER RING COMPRESSOR

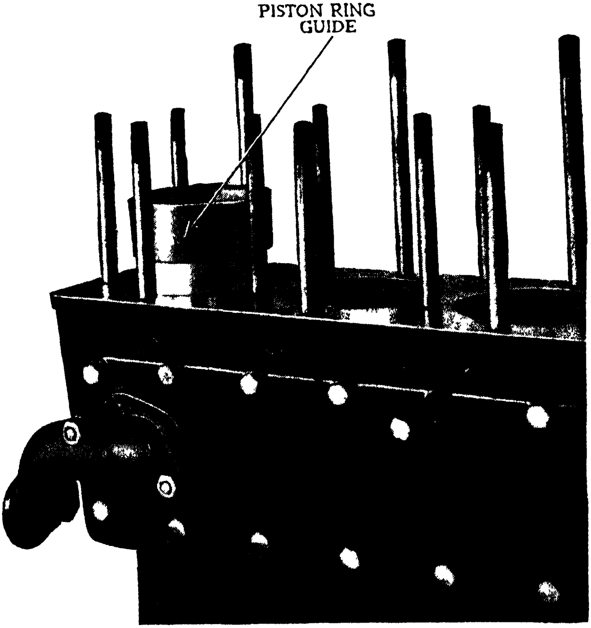
THE GUDGEON PIN

This is also called the wrist pin and the piston pin. There are three ways in which the gudgeon pin is fitted to the piston.

(*a*) Fixed in piston. For small engines the pin is a tapping fit in the piston, one end being a slightly easier fit than the other to facilitate fitting. The pin may be locked by a set-screw through the piston boss. The small end of the connecting rod works on the pin, the rod being bushed for this purpose. This type of fastening is shown in Fig. 20.

(*b*) The pin is fixed to the small end of the connecting rod and works freely in the bosses of the piston, which may be bushed for this purpose. This type is shown in Fig. 10. The fit of the gudgeon pin in the piston must be such that at ordinary room temperature the gudgeon pin can be pushed through the piston with the two thumbs. On no account must the piston holes be reamed out, but new pins, when required, must be selected for the correct fit. This method is good modern practice with alloy pistons.

(*c*) The pin is free in both piston and small end, and this type of pin is called "fully floating." End plugs are used to prevent the pin scoring the cylinder walls, and sometimes the pin is prevented from moving endwise by steel spring rings. Needless to state, this type requires excellent fitting, but is very satisfactory in practice.



(*Courtesy of A.E.C. Ltd.*)

FIG. 19. USE OF GUIDE FOR INSERTING PISTON INTO CYLINDER

The reader is advised to compare the above remarks on the design and fitting of pistons with those given for the larger engines in the section on gas engines (Section XXXII). Every care must be taken in removing and fitting gudgeon pins in these small pistons, as distortion will result if the pin is too tight, apart from seizure under running conditions.

THE BIG ENDS AND CRANK PINS

In the fitting and design of the big ends of the connecting rods the practice has changed considerably in the last few years. The treatment by scraping, etc., as in the case of the larger engines, has been discontinued and a policy of replacement has been adopted. The reason for this is not because the quality of workmanship is not good enough, but because the problems of lubrication are only recently being understood. The reader is advised to consult the Report of the Institution of Automobile Engineers, 1941, on this topic.

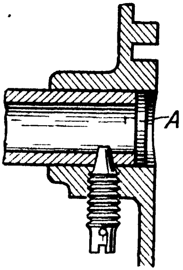


FIG. 20. SECURING GUDGEON PIN IN PISTON

Briefly, the conditions at the big end are these. The bearing surfaces are oiled under pressure from the main supply through the crank pin and there is a continuous flow of oil over the bearing surfaces. The strength of an oil film when very thin is tremendous, and the explosion pressure is not sufficient to break this film and cause metal to metal contact. If the clearance is not enough, the oil film will not be continuous and metal to metal contact will arise. On the other hand, if the clearance is too great, the film will have no strength.

The running temperature of the big ends and crank pin is well above that of boiling water, and the oil is quite "thin" at this temperature. In fact, a thin oil gives a lower temperature at the big end than thick oil. For these reasons there is always a clearance between the connecting rod big end and the crank pin.

Modern construction is demonstrated in Fig. 10 where the big end bearings are formed by die casting the Babbitt metal directly into the rod, and shims are provided for the easy adjustment of the clearance. The bearing surfaces are quite plain except for a "mud-groove" at the joint. The actual clearance depends upon the design and the materials used, and maker's figures should be consulted. A rough and ready method which gives quite good results is to place a cigarette paper on the crank pin or journal and tighten up the bearing with this in position. If the clearance is correct, the connecting rod can be comfortably moved by hand, but certainly would not fall under its own weight. The adjustment must never be made by filing the cap of the bearing, but by means of the shims.

When the big end is being dismantled, the shims must be kept carefully to their correct positions, and not changed about in any way, nor must the caps be reversed or changed. The caps and rods are always marked by the makers to facilitate reassembly.

In reassembling, the big end bolts must not be overstrained, as noted in the general instructions at the commencement of this section, but they must be tight and securely locked. It is always better to throw away the old tab-washers and replace by new ones. If a bolt is strained or damaged in any way, always replace it by a new one; do not attempt to use an ordinary mild steel bolt which may "nearly fit." On all work in connection with the big ends and main bearings, every suspicion of dirt must be excluded, and all parts well brushed in paraffin, dried, and lubricated with good clean oil before reassembly.

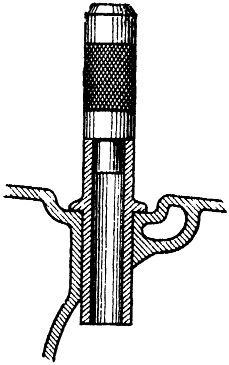


Fig. 21A. Inserting Valve Stem Guide

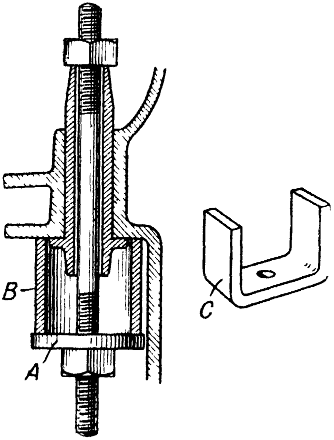


Fig. 21B. Removing Valve Stem Guide

It is beyond the scope of this section to deal with oval crank pins; in fact, even if the tools were available, it would be better to replace the crankshaft by one which has been reground at the works and has the connecting rods fitted, and main bearings also.

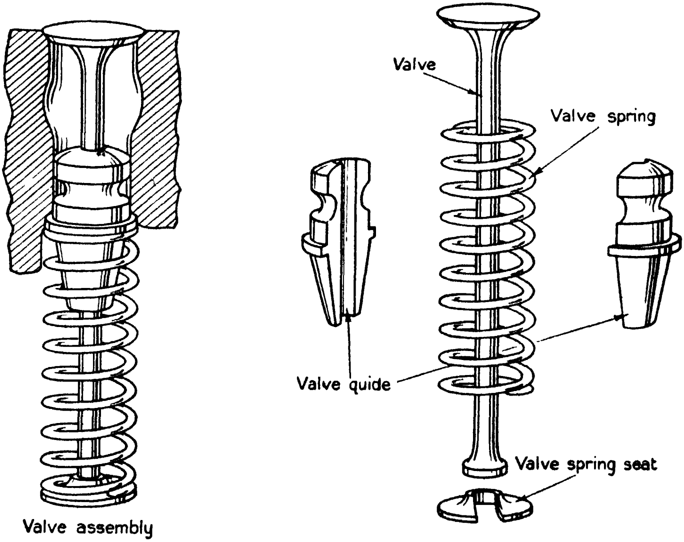
If the crankshaft is removed, it should be stood on end, as seen in Fig. 2, to prevent distortion.

VALVES

The valves used in the petrol engine are of the mushroom or "poppet" type, and the seating is usually 45°. The stems of the valves work in guides which are replaceable. The solid type of valve guide is inserted and removed as shown in Figs. 21A and 21B.

In the case of the Ford 8 engine the valve guides are in two halves which fit together as in Fig. 22. Another example of the solid guide for an overhead valve is shown in Fig. 5.

The inlet valves run cooler than the exhaust valves, and naturally the seating of the valve and the casting on the inlet valve shows less wear and tear. However, the inlet valve stem gives more trouble, as leakage there due to slackness in the guides introduces air and spoils the carburation, especially under slow running conditions; also oil is drawn in, which, when oxidized, becomes gummy and causes sticking.



(*Courtesy of the Ford Motor Co., Ltd.*)

FIG. 22

TREATMENT OF VALVES

Whether overhead valves or side-by-side valves, the general treatment is the same, except that the overhead valve is usually more easily accessible. First the valve must be drawn by compressing the spring (or springs if there are two). There are many variations of valve spring compressors on the market, three being shown in Figs. 24, 25, and 26. (See also Fig. 28.)

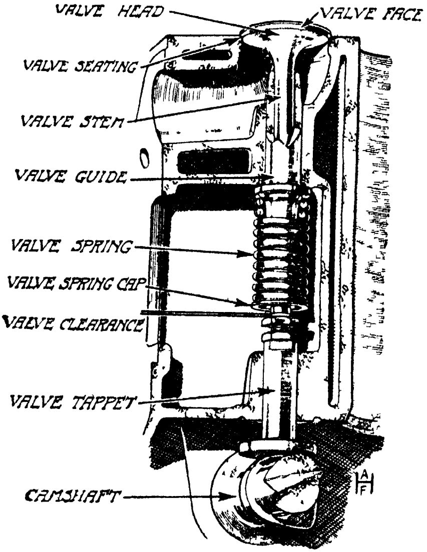


Fig. 23. Side Valve Gear

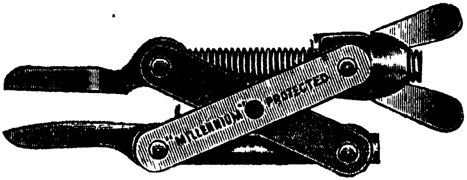


Fig. 24. "Millennium" Valve Spring Compressor

If a compressor is not available, the springs may be compressed by means of screwdriver and packing pieces. This is not to be recommended as it is easy to

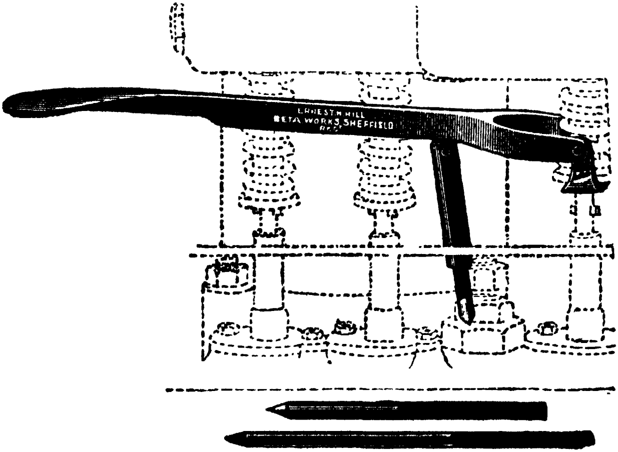


FIG. 25. "NESTHILL" VALVE SPRING COMPRESSOR

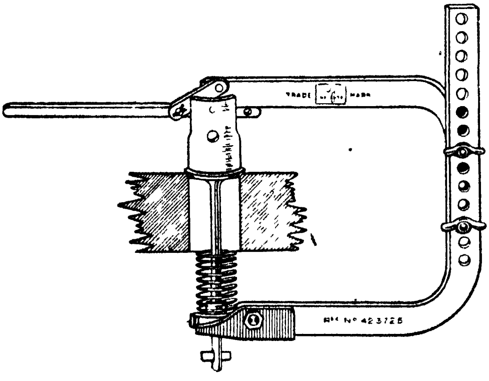
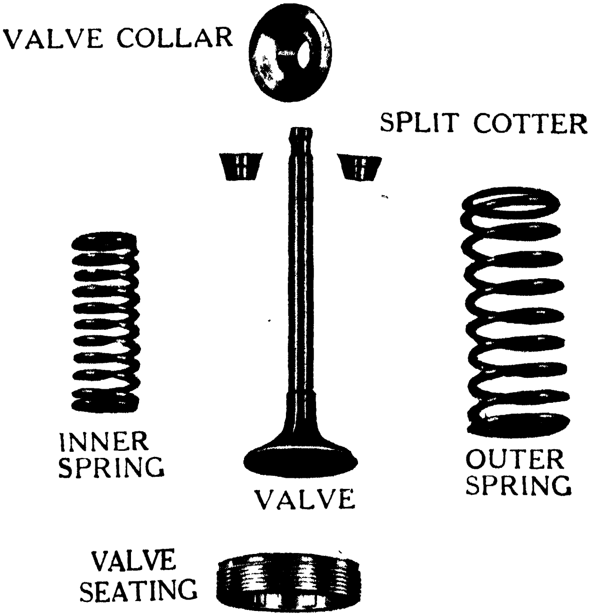


FIG. 26. "INSTANTO" VALVE SPRING COMPRESSOR

trap one's fingers. Moreover, on assembly much time may be wasted in looking for lost parts. In any case, handling the valve springs calls for patience and dexterity. The split cotters, as shown in Fig. 27, are easily lost in reassembling.



(*Courtesy of A.E.C., Ltd.*)

FIG. 27. VALVE PARTS

After removal, the valves and parts are washed clean in paraffin and then carefully examined. Each valve will be found to be marked for its appropriate position, which of course must be retained on assembling.

The valves are made of suitable alloy steel to withstand the difficult conditions under which they work, usually the alloy being of the tungsten-vanadium-chromium variety. However, the seats will be found to be pitted, and perhaps the valves will be found to be warped. If very badly pitted or warped the valve must be replaced, or may be reground in a special machine supplied for the purpose. Next, the seating in the casting must be examined and recut by a tool similar to that in Fig. 30. This recutting requires care or otherwise "chatter" will result, and a great deal of trouble caused in rectifying this again. In Fig. 29, *A* and *C* show the valve and seat as they should be: the width of the seating is not very wide, say $\frac{1}{16}$ in. *B* shows a valve "pocketed" in the seating, and *D* shows a valve which is badly grooved. Neither *B* nor *D* is satisfactory. The seating for the pocketed valve *B* can be recut as at *E*. In good modern practice the valve seat is made of "Stellite" and is removable, as in Fig. 27.

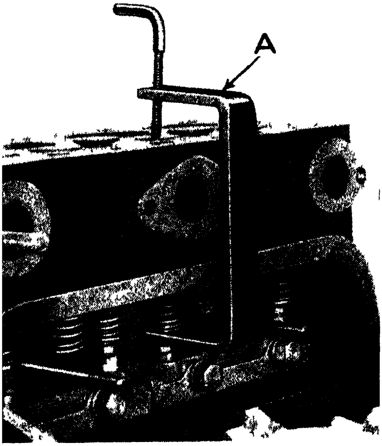


FIG. 28. VALVE SPRING COMPRESSOR

After examination, each valve is tried in its seating by turning it whilst in position. If it is warped, then this can be seen easily, and the seatings will be "marked" where they have rubbed. If a replacement is unnecessary the valve may now be ground in. A light helical spring is placed under the head of the valve as in Fig. 31, sufficient to raise the head clear of the seat. A small amount of medium carborundum paste is applied evenly to the seat and the valve pressed down and rotated backwards and forwards once or twice. Medium pressure gives the best results. The valve is allowed to rise from the seat, a slightly different hold taken on the turning tool, and the rotation repeated. Do not turn the valve round and round without lifting, as this will only result in grooving the valve. If, on examination, the seating on both

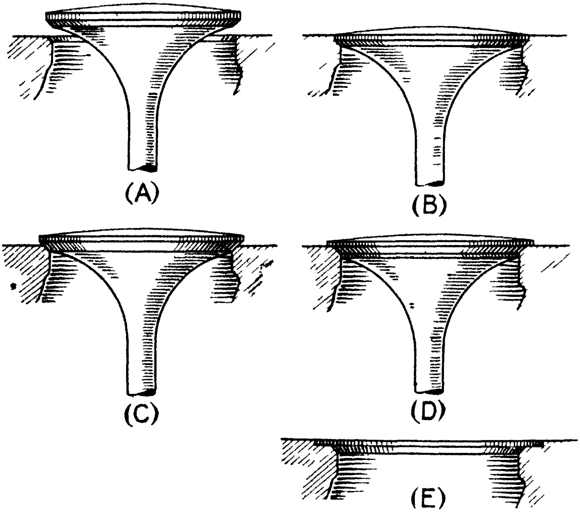


FIG. 29. VALVE WEAR

valve and casting appears to be "coming up," carry on with the method until all the black specks, hollows, etc., are removed. If, however, this appears to be a long process, use a little coarse grinding paste and then revert to the medium. Remove the valve and wash all parts (especially the guide) clean, and see that no

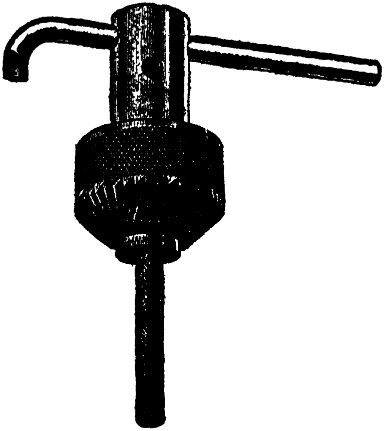


FIG. 30. J. AND S. VALVE RESEATING TOOL

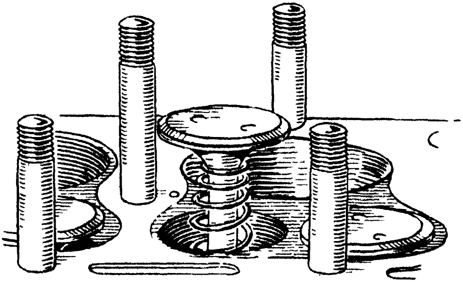


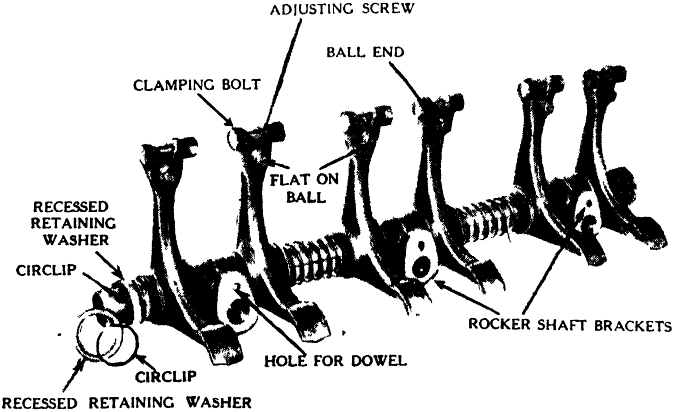
FIG. 31. VALVE GRINDING

abrasive is embedded in the seating. Replace the valve in a dry state (or with a little very fine paste) and finish the seating. The seatings should appear uniform matt surfaces, slightly burnished if the "dry" process has been used. It is not necessary to use excessive quantities of abrasive, and it is as well to see how little can be used, and to watch carefully that none is left behind. The cylinder bores and adjacent valve ports should be plugged with clean rag during the process, as although everything must be carefully cleaned down again before assembly, it is essential to keep the abrasive away from all working parts.

CAMS AND TAPPETS

The overhead camshaft type of engine has the valves operated by bell-cranked levers as in Fig. 32. The assembly is shown in end view in Fig. 5. Other overhead valves are operated by camshafts placed as the one in Fig. 3, the motion being transmitted to rocker arms by means of long push rods. In both these types the adjustment is carried out by a set-screw acting on the end of the valve stem or push rod. In order to make sure that the valve can be seated properly by means of the spring, it is necessary that there be some clearance in the mechanism when the valve is closed. The actual amount of clearance necessary varies with the type of engine and is always measured when all the parts are at normal running temperatures. Too much clearance is better than too little, as if the valve is held slightly off its seat, hot gases will blow by and quickly ruin the seating. Too much clearance will merely cause a little more noise. No attempt should be made to adjust the clearance when newly assembled, but merely a check to see whether there is a reasonable excess of clearance. Then when the engine has been well warmed up and tappets, etc., seated, the clearance can be adjusted to the maker's requirements. As in Fig. 5, this clearance should be tested only when the tappet or rocker is on the heel of the cam, and should be again re-tested after a few hours' run.

In the case of the side-by-side valves, the adjustment is usually accomplished as demonstrated in Fig. 33.



(Courtesy of A.E.C. Ltd.)

FIG. 32. ROCKER SHAFT ASSEMBLY

VALVE TIMING

Owing to the very high speed of operation of the ordinary engine the cams and cam gear have to be carefully designed, and if they are neglected in the matter of adjustment, undue wear takes place which can be rectified only by a new or reground camshaft and new tappets. Since, when the engine is running, the time available is so short for lifting a valve, holding it open, and allowing it to close by means of the spring, the forces called into play are large. If the spring shows signs of losing its strength or temper, which occurs more frequently on the exhaust valves, the spring must be replaced by a new one, as a weak spring will not close the valve rapidly enough. Moreover, a weak spring will allow the valve to chatter on its seat and accelerate wear by erosion due to gas leakage.

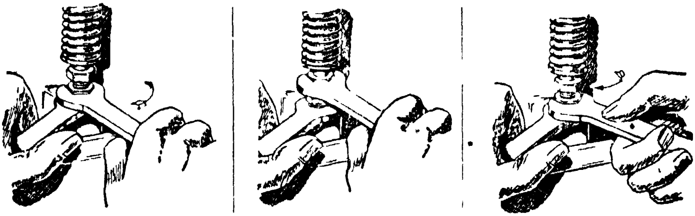


FIG. 33. ADJUSTING MORRIS SIDE VALVE TAPPETS

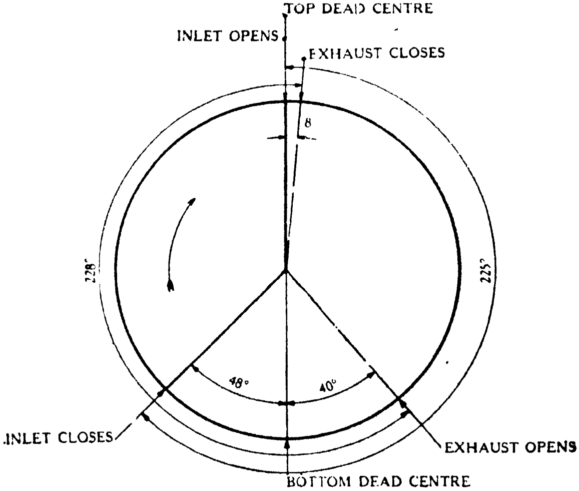
Left. Loosen the lock-nut while holding the tappet by a spanner engaging the flats at its upper end.

Centre. While still holding the tappet, the tappet adjusting screw may be rotated one way or the other to alter the clearance

Right. The lock-nut may now be securely tightened, care being taken that the tappet adjusting screw is not moved relatively to the tappet during this operation

When a valve begins to open, some time elapses before it can be raised to its full lift; in the same way the valve takes time to return to the closed position. This leaves only a very brief period for the valve to remain fully open, and in extra high-speed engines the valve is practically always on the move, with no "dwell" in the fully open position. Also, since the gases flowing from the cylinder, and the charge flowing into the cylinder, take time to get into motion, even when the valve is partly open, it can be seen that the longest period possible should be allowed for valve action.

In this particular case (see Fig. 34), the inlet begins to open at T.D.C., remains open during the whole suction stroke and closes during the compression stroke, 48° after B.D.C. The exhaust opens 40° before B.D.C. on the power stroke, remains open during the exhaust stroke, and closes 8° after T.D.C. There is thus an overlap of 8°, when both valves are open a very small amount. Most engines are timed with a similar overlap, and generally the higher the engine speed the more the overlap and the earlier the inlet valve opens.



(*Courtesy of A.E.C., Ltd.*)

FIG. 34

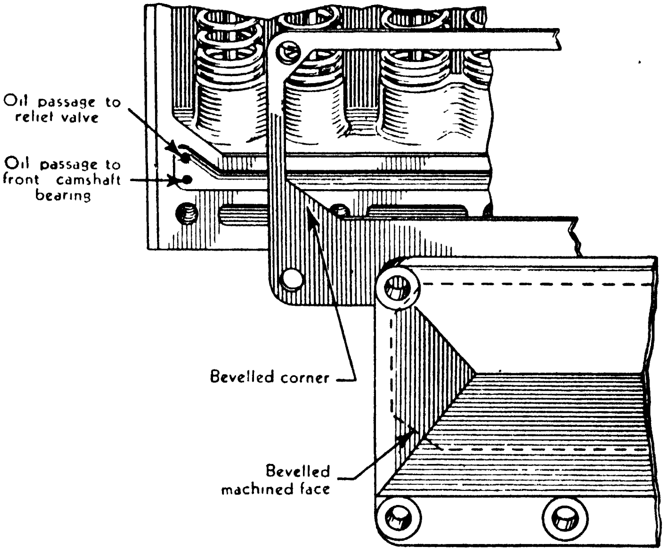
This fact is of great help in assembling an engine when the crankshaft or camshaft has been disturbed. Set the pistons at T.D.C. and B.D.C. Turn the camshaft until this overlap is found by the corresponding inlet and exhaust valves just moving. Then drop the timing gear into position, and on examination of the timing-marks on the wheels they most probably will be correct. If not, then they will not be more than one tooth out, which is easily corrected. For further details on chain adjustment, self-adjusting camshaft drives, etc., the reader must follow the maker's instructions.

JOINTS AND JOINT MAKING

Before assembling two machined surfaces to make a joint, all the old jointing must be scraped off, and the surfaces cleaned. Cleaning is best done by means of a scraper, but care must be taken not to damage the surface. The two surfaces should then be put together to see whether any warping or damage has occurred. The joint may then be made. Copper asbestos gaskets may be used again if in really good order, but they need gentle handling. It is as well to be careful with them even if it is not intended to re-use them, as they may come in as spares if new ones are not available.

The usual jointing compound is gold size for all types of joints. The gold size on being heated dries to a varnish-like surface which later sets hard. If at a later date the joint is "pulled up" again, this surface is disturbed, and the joint may have to be remade. There are compounds on the market which always retain a certain amount of elasticity and permit of this tightening up. Also they leave the surfaces of the joint clean and do not bake themselves on to the surfaces. For oil joints, crank-case joints, etc., often brown paper and gold size are used, and are perfectly satisfactory. A good rule is to use for any joint as thin a jointing material as is consistent with the state of the surfaces. Also care must be taken to see that certain surfaces are left uncovered, or small lubricating holes are not masked. A case in point is shown in Fig. 35.

In tightening up a cylinder head, first make sure that everything is sitting fairly, then bring all the nuts down by hand until they are finger tight. Then fasten down each a little at a time, working on the middle nuts first and trying to equalize the pressure over the whole head. Each nut should be tightened, say, only half a turn at first, so that all the nuts will have been gone over several times before they have all been fully tightened. When the engine has had a run and been well warmed, the nuts may receive a final "pull-up."


Oil passage to relief valve
Oil passage to front camshaft bearing
Bevelled corner
Bevelled machined face

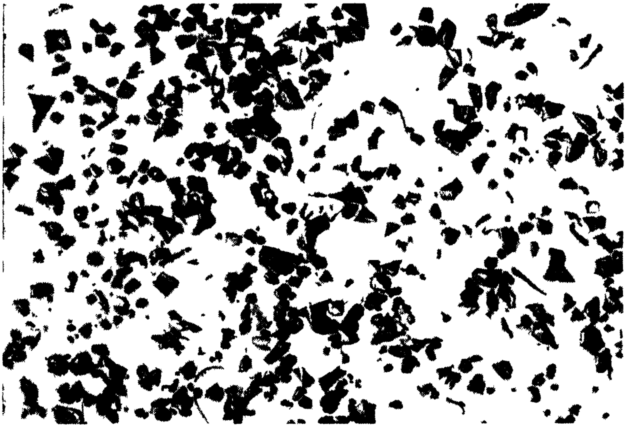
(*Courtesy of the Ford Motor Co., Ltd.*)

FIG. 35

LUBRICATION

The successful maintenance of a petrol engine centres more round efficient lubrication than anything else, and in order to appreciate the problem, the main difficulties must be considered first. The conditions under which the main bearings and crank pins work have already been described, and obviously any failure of the oil supply for a short while will ruin these surfaces. The pistons and cylinder walls are lubricated by means of the oil thrown from the big ends and by the oil vapour present in the crank-case. Larger engines have hollow connecting rods or ducts attached to them so that the gudgeon pin receives the oil under pressure; and a restricted supply may be led to the cylinder walls. The oil on the piston easily becomes partly carbonized with the heat, and forms "carbon"; some is burned in the combustion space, more or less completely. The "oil control" on the piston is accomplished by the scraper ring or rings and general state of the piston and compression rings and their fit in the cylinder. When the cylinder becomes oval special spring compression rings may be used, and they are a surprisingly good palliative, although not as permanent a cure for excessive oil consumption as a rebore. There is a certain amount of "blow-by" past the piston rings when worn, and this dilutes the crank-case oil with the paraffinic content of the petrol and also water vapour which when condensed emulsifies with the oil. This dilution depends largely on the crank-case temperature, and a cold engine shows a greater dilution than a hot one. The paraffin tends to make the oil thin, whereas the emulsified water and oil tends to thicken it. After a few miles, new oil will be diluted by about 10 per cent or so, and this figure will then remain constant. Sometimes the new oil is already diluted before introduction to the crank-case, and in a large fleet this represents a considerable saving.

The partly burned oil causes the colour of the crank-case oil to become black, but the majority of the carbon particles present are extremely small and have little or no effect on the lubricating properties, although they tend to weaken the film. The larger particles are collected at the bottom of the sump. A microscopic view of the particles in the sump oil of a petrol engine is shown in Fig. 36. Roughly between 1 oz. and 2 oz. solids will be gained by a medium-sized petrol engine per 1000 miles running.*



(*Courtesy of the Institution of Automobile Engineers*)

FIG. 36

The paraffinic dilution is not a serious matter, as engines have been run for long periods with over 50 per cent paraffin in the lubricating oil; of course, deliberate dilution to this extent is not to be considered in practice, as the margin of reliability is much reduced. The water and oil emulsion tends to form sludge with the carbon particles.

Therefore, the chief factors to be watched in the matter of the lubricating oil are—

1. A good clean oil of suitable viscosity according to

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* See *I. A. E.*, Vol. IX, No. 2, Nov., 1940.

maker's suggestions, and of the correct chemical stability, etc.

2. Cleanliness. The oil filler must be wiped scrupulously clean of all the oily grit which collects there before filling; see that the crank-case is well flushed of all sludge as already noted.

3. An efficient filtering system continuously in action filtering either the whole of the oil in circulation or by-passing some.

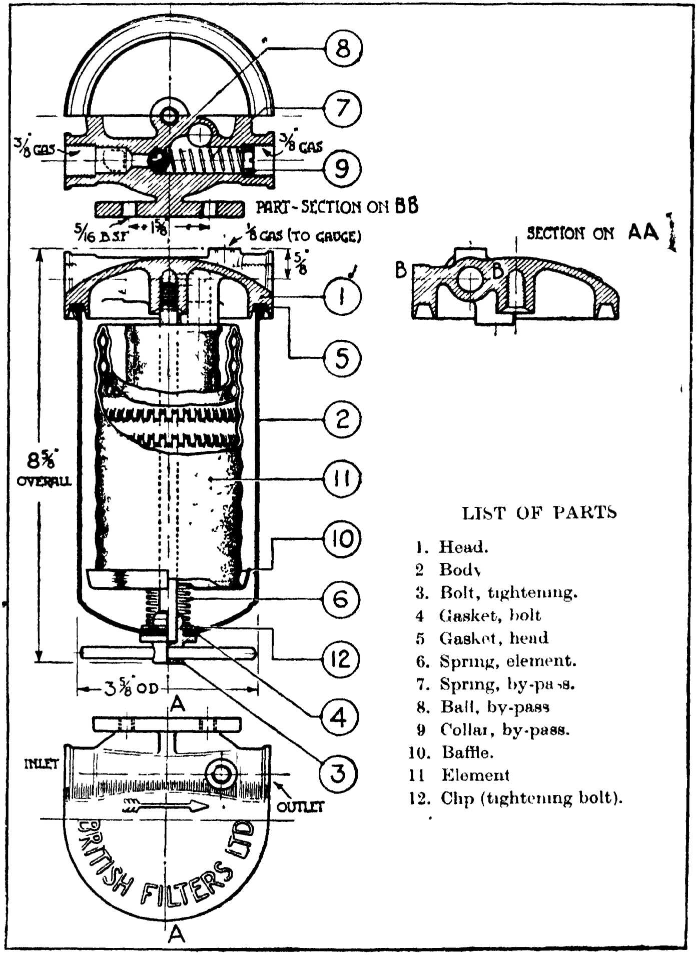
4. An air filter to prevent dust and other particles entering the cylinder.

A typical full-flow oil filter as used by H.M. Ordnance Department is shown in Fig. 37.

Needless to state, any filter employed will need washing out and cleaning periodically and this must be included in the routine adjustments.

With regard to the general engine lubrication the systems vary in their details, and every engine must be considered on its own. The Ford 8 h.p. engine system is shown partly diagrammatically in Fig. 38 and the circuit can be traced. The underside of the engine shown in Fig. 39 illustrates the oil pump and primary system of a larger engine. Notice the two holes on the bottom left, which are the feed and return to the by-pass filter.

The used crank-case oil may be recovered by centrifuging all the sludge from it, or filtering by means of special filters. The recovered oil is practically the colour of new oil, but of course contains the dilution which is normal to the crank-case oil. It is possible to continue this operation repeatedly, as continually new oil must be added to make up the running losses. There is no reason to believe that the recovered oil is inferior in lubricating properties, but there possibly have been some obscure chemical changes taking place whilst the oil was in the engine. Whether the overall

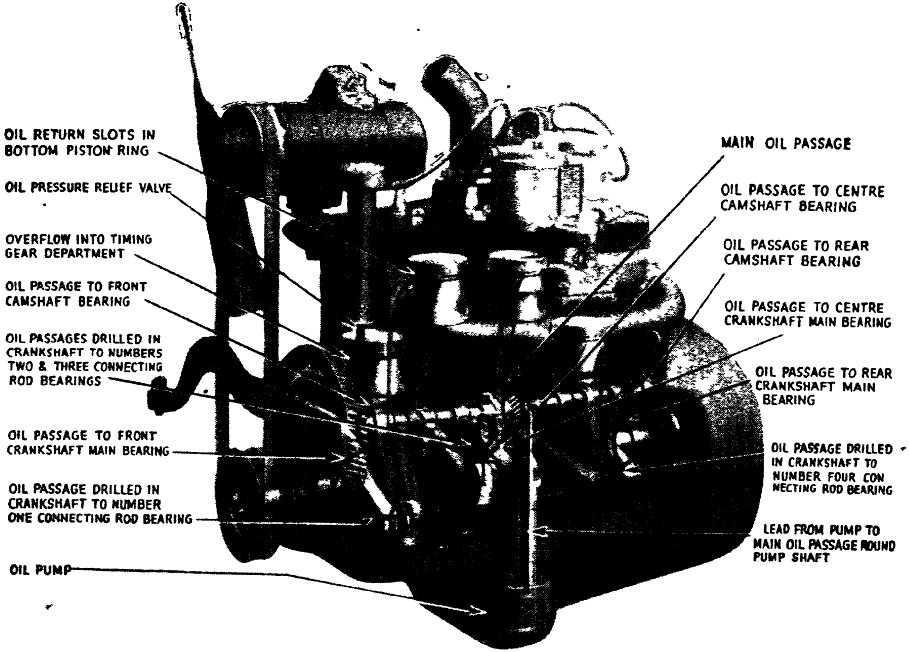


LIST OF PARTS

1. Head.
2. Body
3. Bolt, tightening.
4. Gasket, bolt
5. Gasket, head
6. Spring, element.
7. Spring, by-pass.
8. Ball, by-pass
9. Collar, by-pass.
10. Baffle.
11. Element
12. Clip (tightening bolt).

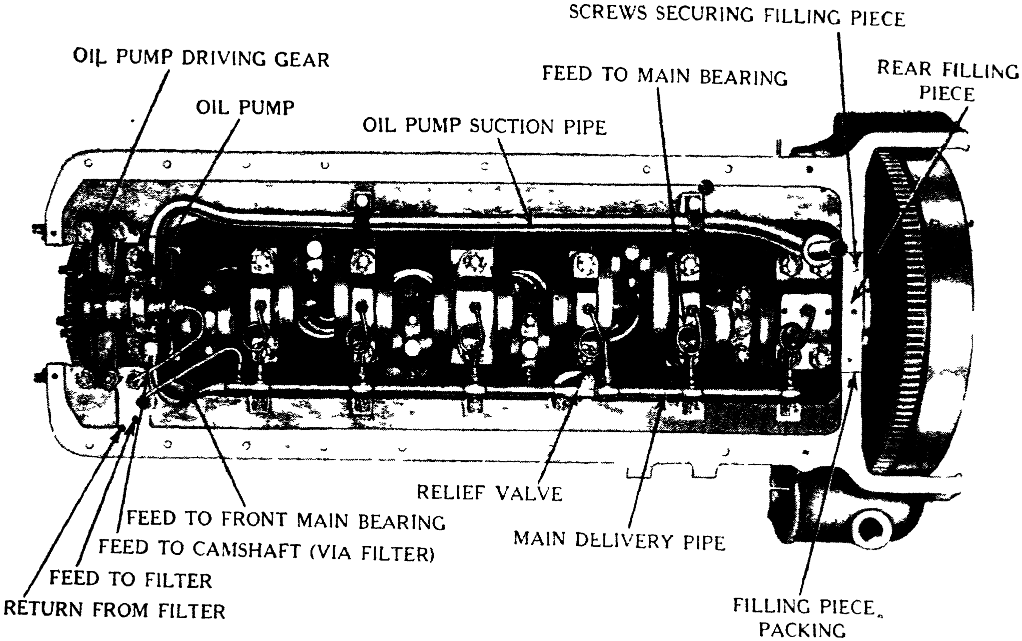
(*Courtesy of the Institution of Automobile Engineers*)

FIG. 37


OIL RETURN SLOTS IN BOTTOM PISTON RING
OIL PRESSURE RELIEF VALVE
OVERFLOW INTO TIMING GEAR DEPARTMENT
OIL PASSAGE TO FRONT CAMSHAFT BEARING
OIL PASSAGES DRILLED IN CRANKSHAFT TO NUMBERS TWO & THREE CONNECTING ROD BEARINGS
OIL PASSAGE TO FRONT CRANKSHAFT MAIN BEARING
OIL PASSAGE DRILLED IN CRANKSHAFT TO NUMBER ONE CONNECTING ROD BEARING
OIL PUMP
MAIN OIL PASSAGE
OIL PASSAGE TO CENTRE CAMSHAFT BEARING
OIL PASSAGE TO REAR CAMSHAFT BEARING
OIL PASSAGE TO CENTRE CRANKSHAFT MAIN BEARING
OIL PASSAGE TO REAR CRANKSHAFT MAIN BEARING
OIL PASSAGE DRILLED IN CRANKSHAFT TO NUMBER FOUR CONNECTING ROD BEARING
LEAD FROM PUMP TO MAIN OIL PASSAGE ROUND PUMP SHAFT

(*Courtesy of the Ford Motor Co., Ltd.*)

FIG. 38. LUBRICATION SYSTEM OF 8 H.P. FORD ENGINE



(*Courtesy of A E C Ltd*)

FIG. 39. UNDERSIDE OF ENGINE SHOWING OIL PIPES, ETC

saving amounts to very much depends entirely on the circumstances, and each case must be considered separately.

If the used oil is decanted into a deep drum fitted with a cover and allowed to stand in a warm place for several months, and then the top oil again decanted, a quite good lubricant results for many unimportant purposes, and it may in case of necessity be used again in the crank-case.

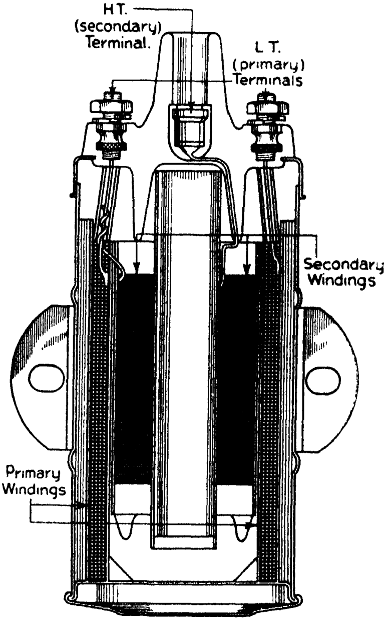
Always pay particular attention to the suction filter as if this becomes clogged the pump loses its suction and the pressure falls in the supply. If a very thick oil is used in cold weather, the same effect may be experienced, and the engine bearings ruined.

Overfilling the crank-case will result in over-oiling the cylinders, and a dirty engine will result, including fouling of the plugs and a smoky exhaust. It is possible to cut down excessive oil consumption in a worn engine by keeping the oil level in the crank-case rather low, but this is obviously a dangerous procedure. Very often a change to an oil of different viscosity will produce a marked reduction in oil consumption.

Over-oiling may also cause sticky valves. This may be temporarily overcome by running the petrol tank low, then adding a little acetone (say ¼ pint to the gallon of petrol). The engine will "pink," but may be humoured on a suitable journey, and most of the gummy deposit will have been removed.

IGNITION

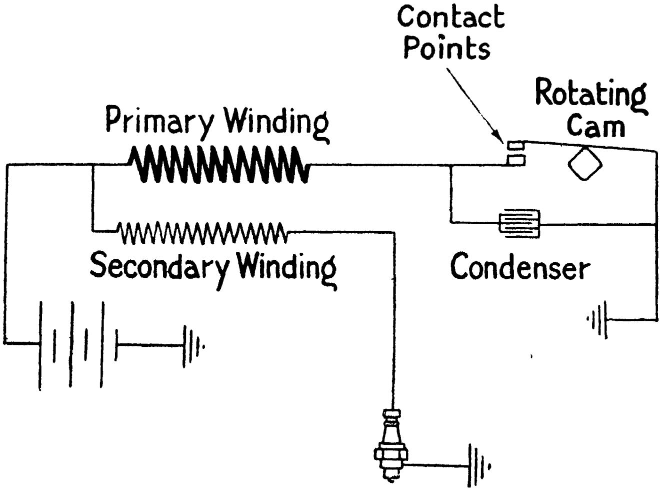
The two chief systems in use to-day are the high-tension non-trembler coil for battery ignition, and the magneto. The former is more common on the smaller engines. The electrical and magnetic principles involved will be dealt with here only in so far as they affect the technique of adjustment and maintenance. First it must be clearly understood that there is no justification whatsoever for the superiority of one system over the other from the point of view of the intensity of the spark. Any variation in the performance of a particular engine when the system is changed is due to altered timing of the spark, and not to its intensity. The burning of the charge in the cylinder is independent of the intensity of the ignition spark, but obviously a "good hot spark" will overcome losses and deficiencies in the system that a weaker one would not.



(*Courtesy of the Ford Motor Co., Ltd.*)

FIG. 40

A section of a high-tension coil is shown in Fig. 40. Just inside the outer cover is the primary winding through which the battery current flows when the contact points in the distributor "make" contact. The low-tension current flowing causes the central core of soft iron wire (the armature) to become highly magnetized. When the contacts "break," this magnetism suddenly collapses, and the lines of force cutting the secondary winding shown between the primary and the armature induce in the secondary winding a very high voltage which is led from the central top terminal to the distributor, thence to the plug.



(*Courtesy of the Ford Motor Co., Ltd.*)

FIG. 41

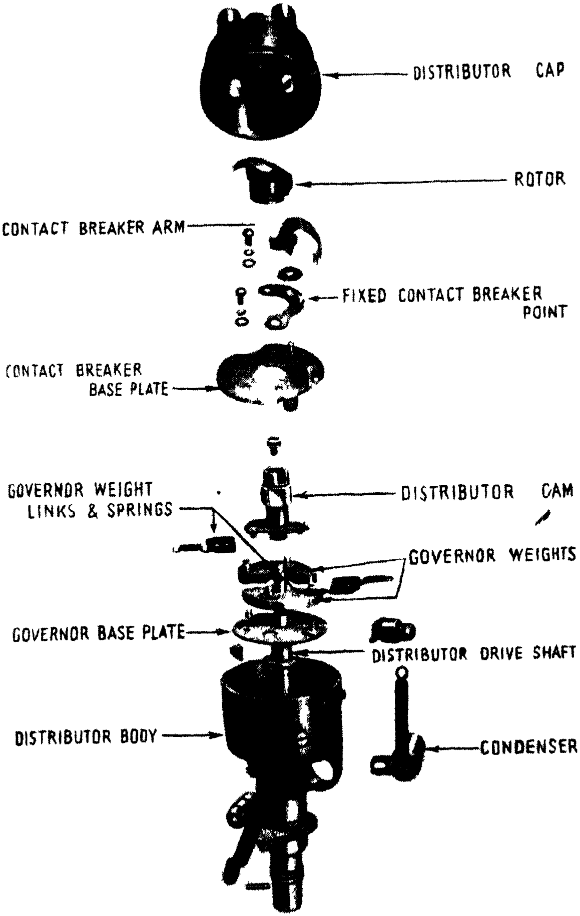
The circuit is shown in diagram form in Fig. 41. A condenser is connected across the contact points to reduce sparking at them. When a coil is considered to be faulty, it is easily tested by trying it connected (intermittently) across the terminals of the battery and arranging a piece of stiff copper wire from the central terminal to form a gap of about $\frac{1}{8}$ in. from the casing. On breaking the battery connection, a good spark should "snap" across the gap. If not, then most likely the secondary winding has broken down and the coil must be replaced by a new one, as the secondary is sealed and the repair is a specialized job. The one coil serves for all the cylinders.

THE DISTRIBUTOR

The make and break contacts, the condenser, the automatic advance, and the high-tension distributor are all housed in one unit, the distributor. The main vertical spindle is driven by screw gears from the camshaft, and when the distributor body is drawn from the engine, the gears are left in mesh, as the drive is carried by a small key or pin seen at the bottom of the exploded view in Fig. 42. On reassembling, the distributor may be put back 180° out of phase in error. This does not matter, as it is easily rectified, but causes needless work in changing the leads.

Probably the most neglected part of the distributor is the automatic advance mechanism. Generally this is not of very robust design, and being neglected, it either does not function at all or, at best, is erratic in action. Often indifferent engine performance is put down to carburation, etc., when it is really the distributor mechanism. As the speed increases, the governor weights fly out against the action of the springs and advance the cam, giving the extra ignition advance necessary for the higher speed.

When the whole distributor has been removed from the engine, wash all parts well with petrol. Reassemble

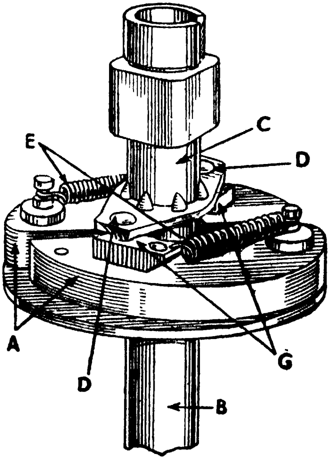


(*Courtesy of the Ford Motor Co., Ltd.*)

FIG. 42. "EXPLODED" VIEW OF DISTRIBUTOR

the governor gear and give all moving parts a little good oil, and keep the oil to these parts only.

When putting back the contact breaker arm, give a trace of oil to the bush and pin, and see that they are a good fit. Too tight a fit will cause erratic running, and too loose is almost as bad. The blade springs should be run through the greasy fingers to supply a trace of lubricant between them, and the heavier gauge guard spring should be inside the curve to prevent undue flexing. See that the points "square up." A trace of vaseline may be given to the cam, then adjust the points. When fully open, the gap should be about 0·01 in. ($\frac{1}{100}$ in.), or a little more. If the contacts are burned, or are not in contact all over their surfaces,



(*Courtesy of the Ford Motor Co., Ltd.*)

FIG. 43. AUTOMATIC ADVANCE MECHANISM

A—Governor weights.
B—Driving spindle.
C—Extension carrying cam.
D, *G*—Pins or links, which when weights fly outwards cause angular advancement of cam *C*
E—Tension springs.

they must be filed carefully, and when finished they should be smooth and clean. The high-tension distributor arm may be slightly burned, but this is of no consequence provided the spark gap is not excessive.

Before putting the low-tension wire back on to the terminal of the distributor, switch on the ignition and flick the wire against the engine casting. This should give a good spark from the high-tension wire of the coil to earth, a suitable gap having been arranged previously. Next see that the contacts are "made" and again flick the wire against the terminal on the distributor. If there is no spark from the high-tension, then the points are not making good contact. Then open the points by turning the engine or putting a piece of paper between them, and touch the terminal again. There should be no spark or at most a very slight one owing to the capacity of the condenser. If

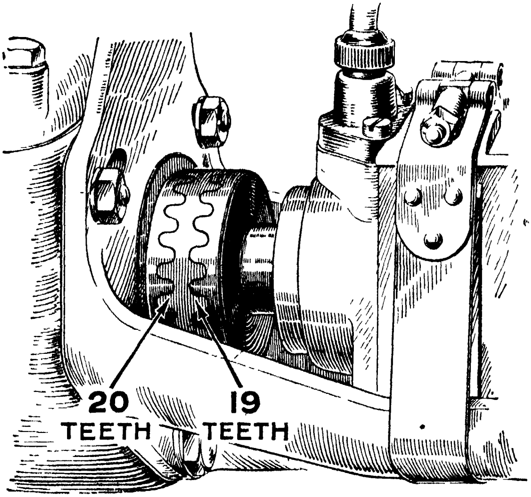


Fig. 44. Magneto Drive Vernier Coupling

there is a good spark, then the condenser insulation has broken down.

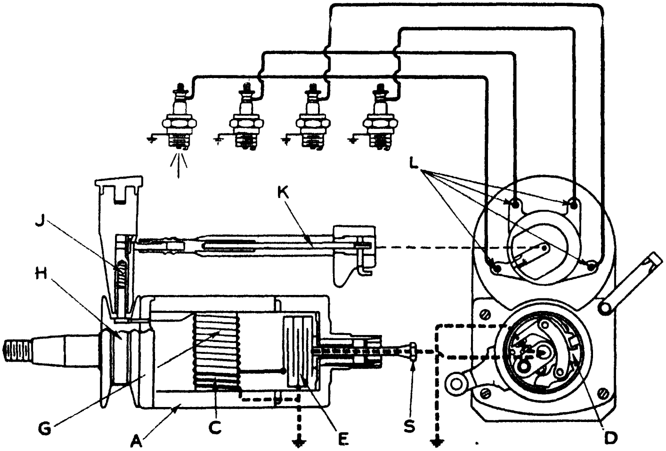
This is often the cause of failure on the road, and if the condenser is removed completely it is possible to get home, perhaps by stopping to clean the points when necessary.

If the high-tension wires have been marked as previously suggested they may be connected to the plugs. However, it is better to check up by referring to the order in which the engine fires. This is done by watching the order in which the valves work, starting from No. 1 cylinder. Find when the two valves for No. 1 cylinder are just opening and closing together; this is nearly the top dead centre of the engine. Then give one complete turn which brings the engine to firing point for this cylinder. The contact points should be just breaking. Then following the rotation of the distributor, the other cylinders can be connected in firing order.

When the engine is ready for the road the final adjustment can be given to the ignition. Bring the ignition forward by turning the whole distributor against the direction of rotation until "pinking" occurs, then slightly retard from this point. One or two runs when the engine is hot will find this position of "incipient knock," which is the correct ignition timing for the particular fuel, mixture-strength, engine temperature, etc. If the governor automatic advance is not suitable for the engine, then a compromise must be struck as dictated by the tendency to knock or harshness of running over the full range of speed and throttle.

THE MAGNETO

When dismantling the magneto, the greatest care must be taken, as not only must the magneto be a good electrical job, but it must primarily be of the highest mechanical reliability. The circuit as shown in Fig. 45 is fundamentally the same as previously dealt with in battery and coil ignition. The primary current (which is broken by the contact points) is induced by the rotation of the armature between the poles of a permanent magnet. In removing the armature, do not omit to remove the high-tension brush *J*, as the shrouds of the high-tension slip ring *H* are easily broken.



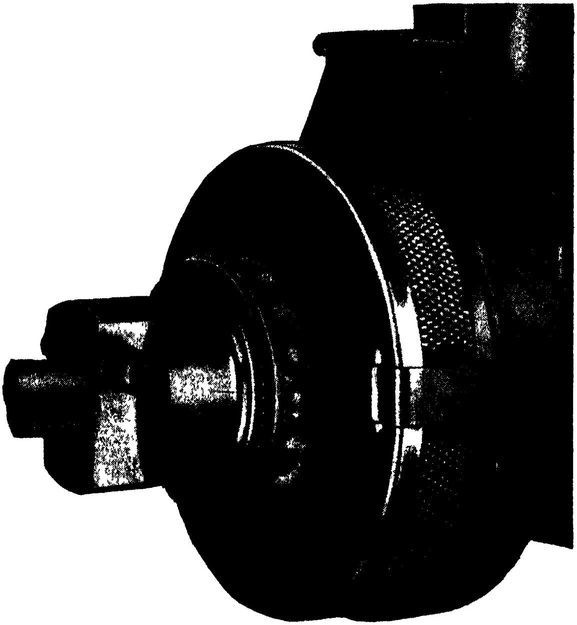
(*Courtesy of the Ford Motor Co., Ltd.*)

FIG. 45

A Armature, *C* Primary winding, *G* Secondary winding, *E* Condenser, *H* and *J* High-tension slip ring and brush, *K* Pencil to distributor arm.

With regard to the contact-breaker, the same general instructions apply as given previously. Here, the contact-breaker assembly is much more robust as the whole assembly rotates. The advance and retard is provided by rotation of the cam ring *D*. The actual timing of the magneto is often accomplished by a vernier coupling as in Fig. 46.

A very satisfactory conversion from coil to magneto ignition is provided by the Scintilla "Vertex" magneto,



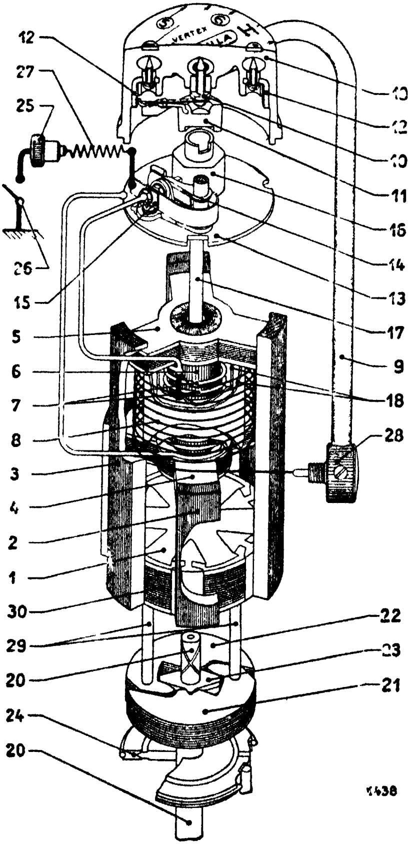
(*Courtesy of A.E.C., Ltd.*)

FIG. 46. VERNIER COUPLING

which includes in its construction an automatic ignition advance of a robust nature. The coil distributor is removed and the Vertex is fitted in its place. An explanatory diagram of this magneto is shown in Fig. 47.

SPARKING PLUGS

The sparking plug works under difficult conditions, and it is important that the correct plug for the particular engine be used as recommended by the manufacturers. Apart from the "long" or "short"



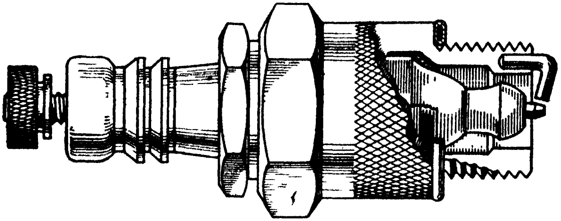
1. Rotating permanent magnet.
2. Short pole shoe.
3. Long pole shoe.
4. Core connecting piece.
5. Core connecting piece.
6. Armature core.
7. Primary winding.
8. Secondary winding.
9. High-tension distributor lead.
10. Central contact in distributor head.
11. Distributor rotor.
12. Electrodes in distributor head.
13. Contact-breaker base plate.
14. Contact-breaker rocker arm.
15. Fixed contact point carrier.
16. Cam.
17. Rotor spindle.
18. Condenser.
19. Distributor head.
20. Driving spindle with brake.
21. }
22. } Centrifugal weights for the automatic advance mechanism.
23. Cam for the automatic advance mechanism.
24. Friction brake.
25. Short circuiting terminal.
26. Ignition switch.
27. Resistance.
28. High-tension coil terminal.
29. Guide pins for centrifugal weights.
30. Laminated rotor.

(Courtesy of Scintilla, Ltd.)

FIG. 47. "VERTEX" MAGNETO

reach of the plug into the cylinder, the central electrode and that part of the insulator inside the cylinder must

be kept at the correct temperature for efficient working. Broadly, a hot engine needs a stout massive insulator round the electrode to keep it cool, otherwise the electrode will overheat and burn itself away quickly and the surface of the insulator be spoiled. If on inspection of the plug the points are burned away and the interior of the plug looks reddish-yellow and powdery, then the plug has been running at too high a temperature. (It may be that the mixture has been weak, causing a long slow-burning combustion, which not only ruins the plugs but also the exhaust valves. The saving on fuel may not be justified in this case.) However, if the plug interior is black and perhaps oily, then the plug is too cold for the particular engine. This condition often causes misfiring due to leakage over the carbon deposited on the insulator, and starting may be difficult. A higher temperature keeps the carbon from being deposited. In any case, it is necessary to adjust and clean the plugs periodically, and not wait until trouble develops.



(*Courtesy of the Ford Motor Co., Ltd.*)

FIG. 48

The insulator may be mica, steatite or silimanite. The latter are minerals and are compressed and hardened to shape.

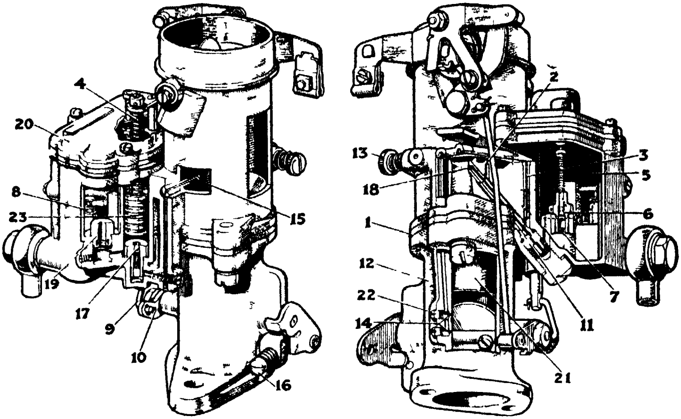
The plug neck ring (the smaller nut) should be gently held in the vice and the plug body unscrewed a little way. Then the parts should be taken apart, due care being taken to prevent injury to the insulator. There is a small washer usually of copper just under the neck ring, and another between the insulator and the body. The latter may be seen in Fig. 48. Trim up carefully the central electrode; do not attempt to bend it, as this will crack the insulator. Carefully scrape clean the insulator and polish it with a piece of fine emery. This removes carbon and other foreign matter which may have embedded itself in the surface of the electrode. Carefully reassemble, seeing that all parts sit fairly before tightening the neck bush. If the insulator is cracked the plug will be ruined, and the above remarks apply only to plugs which are made by one of the reputable manufacturers. Some plugs have porcelain insulators and no screwed neck ring, and it is not possible to dissemble them for cleaning.

Next, adjust the spark gap by bending inwards the electrode or electrodes which are attached to the body. Do not attempt to adjust the central electrode. The gap should be about 0·015 in. to 0·020 in. Too large a gap will throw unnecessary strain on the electrical system and cause difficult starting.

The interference suppressors used for wireless act adversely on the plug performance, and when installed the plugs should be tried "sparking in air." When the plugs are working under compression pressure, however, the resultant potential may be insufficient to cause them to spark. If there is any doubt, the suppressors may be short-circuited temporarily.

THE CARBURETTOR

The types and varieties of carburettors which are in modern use are so many that they cannot be treated separately in this section. The modern carburettor is an exceedingly complex piece of mechanism, and, in the author's opinion, is over-developed; much can be said for simplicity in spite of the relatively poor advertising power. The main requirements of a carburettor will be set out to enable the reader to trace how these requirements are met by the particular carburettor on which he may be working.



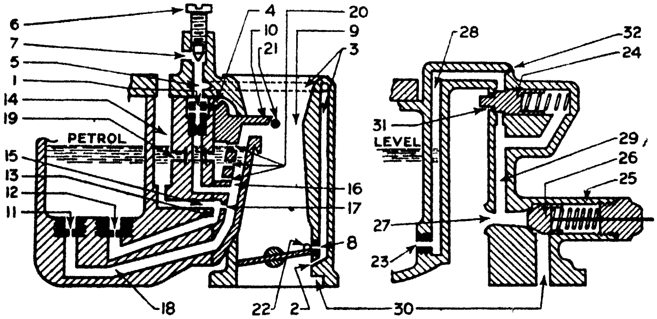
(*Courtesy of the Zenith Carburettor Co., Ltd.*)

FIG. 49. BENDIX-STROMBERG CARBURETTOR TYPES DA 36 AND DA 42 (DOWN-DRAUGHT)

1. Main Body Gasket
2. Air Bleed
3. Idle Tube
4. Pump Spring
5. Float
6. Economizer Valve
7. By-pass Jet
8. Needle Valve.
9. Pump Link
10. Holes-Pump Lever
11. Metering Jet.
12. Throttle Valve Body.
13. Idling Air Screw
14. Idle Discharge Holes
15. Pump Discharge Jet
16. Throttle Stop Screw
17. Delivery Ball Valve
18. Main Discharge Jet
19. Check Disc Valve
20. Float Chamber Cover
21. Venturi Tube
22. Plugs
23. Pump Piston.

1. The carburettor is a measuring device, measuring a correct amount of fuel to suit the air flow. If vaporization can be accomplished partly, and atomization completely, so much the better. But it must be remembered that vaporization requires heat, atomization does not.

2. Easy starting and slow running. Both of these need an extra rich mixture, but not by flooding the carburettor. Usually a separate small carburettor is involved. The throttle is closed, or all but closed, in this position and the main engine suction is by-passed through this small carburettor. Control may be effected by an "air bleed" screw.



(*Courtesy of the Ford Motor Co., Ltd.*)

Fig. 50

1. Slow running jet. 7. Air bleed. 11. Main jet. 12. Compensator jet. 24. Automatic air valve. 25. Starting control valve.

3. A reasonably constant mixture-strength must be maintained over the whole range of "pulling" at all speeds. A simple jet in a choke unfortunately gives too weak a mixture at low air speeds and too rich at high speeds. Most carburettors include a device for correcting this, by compensator jets or drowned jets.

4. Good acceleration and immediate pick-up must be ensured by a reserve of fuel which can be immediately supplied to the air from a reservoir or by means of a pump.

5. A weak mixture coupled with extra ignition advance for "cruising."

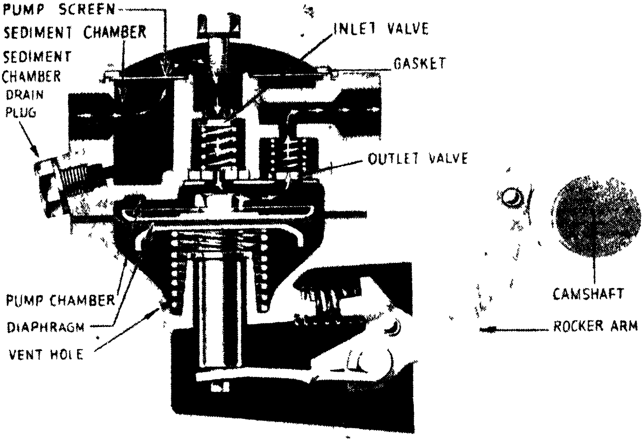
The above broad outline will be helpful in adjusting the carburettor.

DISMANTLING THE CARBURETTOR

Before removing any of the controls, examine the position of throttle, etc., carefully, as this will facilitate reassembly. When removing the petrol pipe, see that the petrol cannot siphon over from the tank. This is not likely, but may happen in certain circumstances. Having removed the carburettor, take out the float carefully and tip the petrol in the float chamber into a clean vessel. If there is any water or dirt it will be immediately visible, and steps must be taken to see that the petrol filters are undamaged. Then remove the caps to the various jets and examine the contents in the same way. Thoroughly clean all ducts and passages (a pipe cleaner comes in useful, but make sure that no "fluff" is left behind). Then thoroughly wash all out in petrol and blow the passages dry (using a piece of rubber tubing). Needless to say, do this well away from any flame. Examine the jets carefully and see that they are clean. Do not attempt to force wire through them or otherwise enlarge them, as they have been calibrated. Nothing is more annoying than to find a jet larger than the size stamped on it. Then reassemble all parts in their correct position, and see that jet washers are exactly as they were before dismantling. Pick the small hairs, etc., from the filters as far as possible before brushing the filters in petrol. The filters are not easy to clean. See that the needle valve sits properly on its seat. It may be necessary to tap this lightly with a hammer to get a good seat. Lastly, see that the float is right way up when inserted in the float-chamber. When connected up, test the carburettor to see that it does not flood.

ADJUSTMENT OF CARBURETTOR

Here, the previous adjustment must be recalled and the maker's instructions followed. On reassembly, especially if the whole engine has been dismantled, the immediate results are not very encouraging, as everything needs some adjustment or other. A little patience, and a desire to see that the engine is not swamped in petrol, will achieve much. Do not attempt to get good slow running at first, but let the engine run at a moderate speed.



(*Courtesy of the Ford Motor Co., Ltd.*)

FIG. 51. MECHANICAL FUEL PUMP

When the engine is on the road, the final adjustments to the carburettor may be made. It is not always easy to distinguish between carburettor faults and ignition faults, one being blamed for the other, but the following chief points may be borne in mind.

1. A weak mixture will cause "popping back" in the carburettor, bad pulling power, and a hot engine. If the carburettor setting of jets, etc., is correct, then it may be there is an air leak due to a bad joint on the manifold. A lighted match held near the joints will indicate the leakage, care being taken to keep the flame away from the carburettor. If the petrol does catch fire due to "popping back," shut off the fuel and race the engine to consume the petrol in the float-chamber. The procedure necessary in case of fire should always be decided *beforehand*. Often the weak mixture will be found to become stronger as the speed rises. According to the type of carburettor, the correction for this will be found in the manual.

2. Too rich a mixture will cause overheating, and blackish smoke from the exhaust, as distinct from the bluish-grey smoke due to lubricating oil. If the carburettor setting is correct, before altering the sizes of jets see that the petrol level is not too high in the float chamber, or that jet washers have not been omitted.

3. Slow running. If good slow running cannot be attained by adjustment, then the fault lies either with the ignition or with air leaks around worn valve stems or round the throttle spindle. A great deal of working takes place at this spindle, and wear, under the conditions of bad lubrication which prevail, is not easy to rectify, except by bushing the holes in the carburettor body.

In any case, when adjustments have been made on the above lines a perusal of the daily log should be made to check up on the consumption. Lastly, there is a great deal in securing the co-operation of the driver. He can help the maintenance workshop considerably and if his interest can be encouraged he can go a long way towards maintaining his engine in good condition.

Of course, too much interference on his part may be harmful, but if he can be shown in the workshop the result of his interference, this eagerness can be directed to more helpful channels. Just as the best engine needs "humouring" for best results, so does the human being, a fact which is often overlooked in these days of high efficiency.

SECTION XXXII

GAS ENGINE FITTING AND ASSEMBLY

BY

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REVISED BY

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SECTION XXXII

GAS ENGINE FITTING AND ASSEMBLY

INTRODUCTORY

Gas engines in general differ from petrol engines in size, and in being of more substantial construction. Speeds are lower and the dimensions of each cylinder are greater, while absolute reliability over long periods of continuous running is essential.

There is much diversity of types and designs. For the majority of gas engines, the four-stroke cycle is adopted and the cylinders lie horizontally. Vertical cylinder engines with various numbers of cylinders working on a common crankshaft and forming a complete unit are, however, often employed.

From the fitting and assembly point of view, much that has been written in the previous section in connection with petrol engines is applicable also to gas engines. Attention will therefore be directed chiefly in this section to features where differences occur.

Single-cylinder gas engines are seldom made in larger sizes than about 150 h.p. Greater powers are obtained by multiplying the number of cylinders, various horizontal, tandem, or vertical arrangements being utilized.

There is little demand at the present day for engines using the ordinary town coal gas in view of the cheaper running costs of heavy oil engines. The demand is, however, still maintained for gas engines using blast furnace gas and suction gas produced from refuse of various kinds. Oil engines for oil producing districts are also often supplied with a breech end adapted to use natural gas until the supply is exhausted, after which a breech end suitable for oil is fitted to the engine. In the smaller sizes used, for instance, for small electric lighting installations, the engines are made readily adaptable for the use of coal gas, petrol, or paraffin.

As mentioned in connection with petrol engines, modern manufacturing methods aim at the elimination of hand fitting by careful and accurate machining. This aim has been very largely but not fully achieved ; it has however, had some influence on repair work also, since it becomes cheaper to use manufacturers' replacements made accurately to gauge, than to make and fit new parts.

Engines are assembled and run under test by the makers, and are then partly dissembled and the parts packed separately for transport ; for instance, pistons, flywheels, connecting rods, crankshafts, or balance weights may be detached and packed separately as required from consideration of size and weight. Many makers stock quantities of parts, so that spares can be readily supplied or engines can be assembled, tested, and dispatched at short notice.

Attention will be directed primarily to what may be described as the ordinary type of gas engine, that is, single cylinder gas engines which, in the larger sizes, require overhead tackle for handling the principal parts.

A general view of such an engine made by Crossley Brothers is shown in Fig. 1, and sectional elevations in Figs. 2 and 2A. Details of this engine are referred to in what follows to illustrate the practice described.

ENGINE ROOM

The environment of an engine is a matter of the greatest importance, and accordingly a typical model

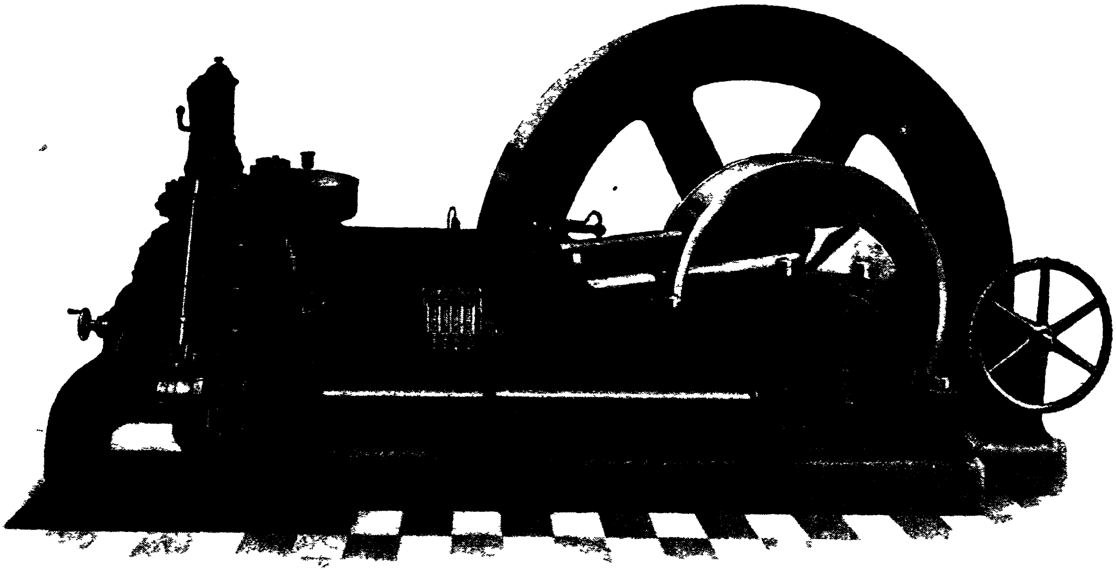


Fig. 1. Crossley Horizontal Gas Engine

engine room is illustrated first in Figs. 3 and 4. The engine must be arranged with ample room all round it, and with its various appurtenances conveniently disposed. It is essential that the engine room should be well ventilated, particularly where suction gas is in use, since it does not smell as strongly as town gas and hence gives less warning, while it is likely to contain more dangerous components or impurities. The various parts and accessories shown will be referred to, as may be necessary, in the following pages.

Vertically over the axis of the crankshaft of the cylinders are arranged two substantial rolled steel joists *B*, each adapted to carry lifting tackle for the erection of the engine in the first instance, and for repair or maintenance work at a later stage. The running gear (not shown) in the case of the transverse or crankshaft girder should run on the lower flanges, so that the upper girder does not prevent a straight run through. The girders and the lifting tackle must be at least strong enough to lift the heaviest engine elements. The gas pipe *A* is connected at some point in its length to a gas bag *C*, having one side of flexible rubber, the object being to make the flow of gas through the meter fairly constant since the suction of the engine is intermittent. Gas engines may be adapted to be worked by petrol, a fuel tank *H* and a carburettor, with a connecting pipe between the two, being then provided.

FOUNDATIONS

The foundations must be substantial and built on firm solid ground. If possible, it is desirable to go down through made ground to the undisturbed subsoil. If the ground is of a yielding character, the foundation should be extended, but preferably a new site should be found. The concrete should be made of 1 part of

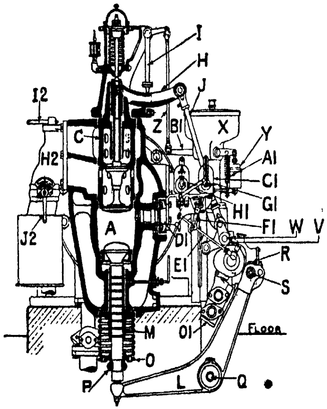


FIG. 2

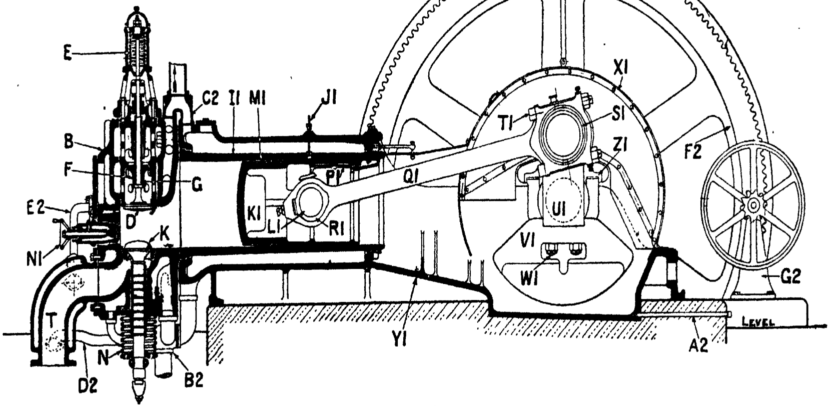
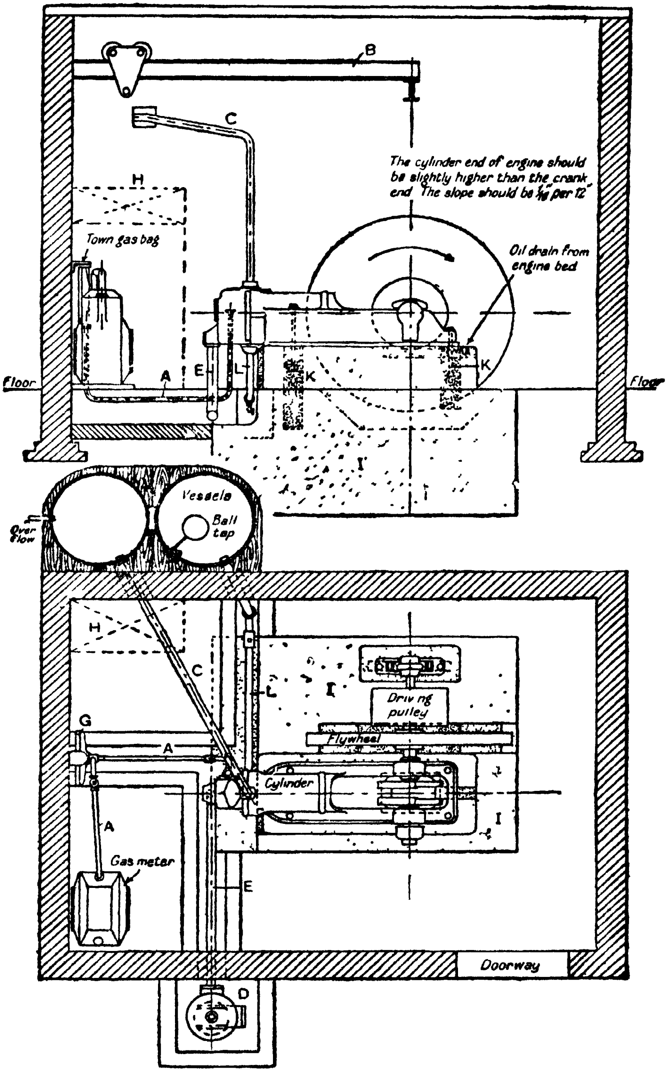


FIG. 2A. SECTIONAL VIEWS OF CROSSLEY HORIZONTAL GAS ENGINE

A.	Combustion chamber
B.	Breech end
C.	Admission valve block
D.	Admission valve
E.	Admission valve spring
F.	Gas valve
G.	Gas valve spring
H.	Radius lever
I.	Fulcrum lever
J.	Admission valve rod
K.	Exhaust valve
L.	" " lever
M.	" " guide
N.	Exhaust valve spring
O.	" " spring plate
P.	" " clip
Q.	" " lever fulcrum pin
R.	" " roller
S.	" " roller pin
T.	Exhaust bend
U.	Combined cam
V.	Admission valve roller
W.	Admission valve pin
X.	Centrifugal governor
Y.	Governor dashpot
Z.	Governor coupling rod
*A*1.	Governor speeder spring
*B*1.	Governor lever
*C*1.	Magnetos
*D*1.	Igniter block
*E*1.	Ignition pusher rod
*F*1.	Ignition timing handle
*G*1.	Magneto lever
*H*1.	Magneto coupling rod
*I*1.	Cylinder liner
*J*1.	Cylinder liner oil tube
*K*1.	Piston
*L*1.	Piston pin
*M*1.	Piston rings
*N*1.	Starter valve
*O*1.	Controller valve
*P*1.	Piston pin lubricator
*Q*1.	Connecting rod
*R*1.	Connecting rod, small end bearings
*S*1.	Connecting rod, large end bearings
*T*1.	Connecting rod bolts
*U*1.	Crankshaft
*V*1.	Balance weights
*W*1.	Balance weights studs
*X*1.	Splasher guard
*Y*1.	Bed plate
*Z*1.	Main bearing caps
*A*2.	Waste oil drain pipe
*B*2.	Water inlet branch
*C*2.	Water outlet branch
*D*2.	Auxiliary water inlet
*E*2.	Auxiliary water outlet
*F*2.	Flywheel
*G*2.	Barring gear
*H*2.	Gas and air block
*I*2.	Handle for regulating gas
*J*2.	Handle for regulating air


B
C
The cylinder end of engine should be slightly higher than the crank end The slope should be 1/16" per 12"
H
Town gas bag
Oil drain from engine bed
Floor
A
E
L
K
K
Floor
I
Vessels
Ball tap
Over flow
H
C
L
I
G
A
Driving pulley
Flywheel
Cylinder
I
A
Gas meter
E
Doorway
D

FIGS. 3 AND 4. TYPICAL ENGINE ROOM

cement to 3 parts of stones or broken bricks, sufficiently small to pass through a 1½ in. mesh, while 2 to 3 parts of clean, sharp sand to 1 of cement should be added. Brickwork should not be used. The concrete must be set hard before use.

The top of the concrete foundation should be left rough, and the engine built up on iron packing pieces, about 3 in. × 3 in. × ½ in. or larger according to the size of engine. The engine should be levelled carefully, so that it is slightly inclined downwards towards the crankshaft end, to facilitate drainage of used oil from the cylinder, and it should be set correctly with respect to the driven machinery. The foundation bolts should

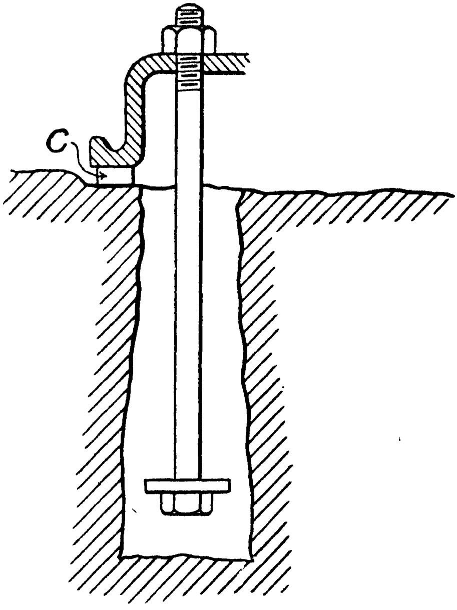


Fig. 5. Holding Down Bolt

be suspended in position in the bolt holes by means of their nuts, as shown in Fig. 5, the holes being left tapered and rough so that the cement which is filled in later will grip well.

The iron packing pieces *C* must give a flat wide bearing area, and should be arranged near the bolt holes. They should be sufficient to support the engine independently of the grouting which should be regarded as a filling. Wedging up by chisels is quite inadequate owing to the very small bearing area. Care must be taken to ensure that all packing pieces share the weight equally so that the engine frame is not twisted or distorted. Any extra thickness necessary should be obtained by pieces of sheet metal giving the same supporting area as the blocks.

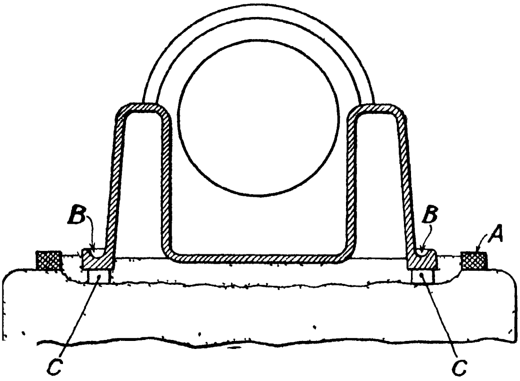


FIG. 6 SECTION OF ENGINE FOUNDATIONS

A clay ridge *A* to retain the grouting, Fig. 6, should be made round the engine bed parallel to the oil groove *B* round the edge of the casting. A board may be used instead of the clay. The bolt holes should then be filled with cement, and cement grouting should be run in level with the oil groove preferably from the inside through holes in the casting, when such are provided for this purpose. By this method it is possible to see whether the grouting has completely filled in the space all round the bed. When the cement has set firmly, the engine bed plate will be uniformly supported all round, and the nuts on the foundation bolts may be tightened up. Where the engine bed is unsupported under the foundation bolt as in Fig. 5, do not over-tighten the nut, as this may distort the bed. The oil groove *B* is provided to prevent damage to the concrete, but as a further precaution the concrete may be painted.

When the grouting has partly set, the clay ridge or board may be removed, and the cement rounded or filled up and generally neatly finished. The soleplate under the independent third crankshaft bearing must be similarly mounted on a concrete foundation and secured in position.

The trenches for the exhaust pipes, etc., should be brick lined, and covered with removable iron plates level with the floor. They should be well drained, and give room for remaking joints, etc.

CRANKSHAFT ALIGNMENT

In small engines the flywheel, or flywheels, are overhung on one or both sides of the two bearings carried by the bed plate ; but in larger engines a third bearing is provided outside the flywheel and driving pulley. The driving pulley is, however, often arranged outside the third bearing, this arrangement as applied to Crossley engines being shown in Fig. 7. It is essential that all the bearings should be truly in line. Any non-alignment brings about continual bending of the shaft as it rotates. This results in heating and, ultimately, in seizing of the bearings, or in fracture of the shaft.

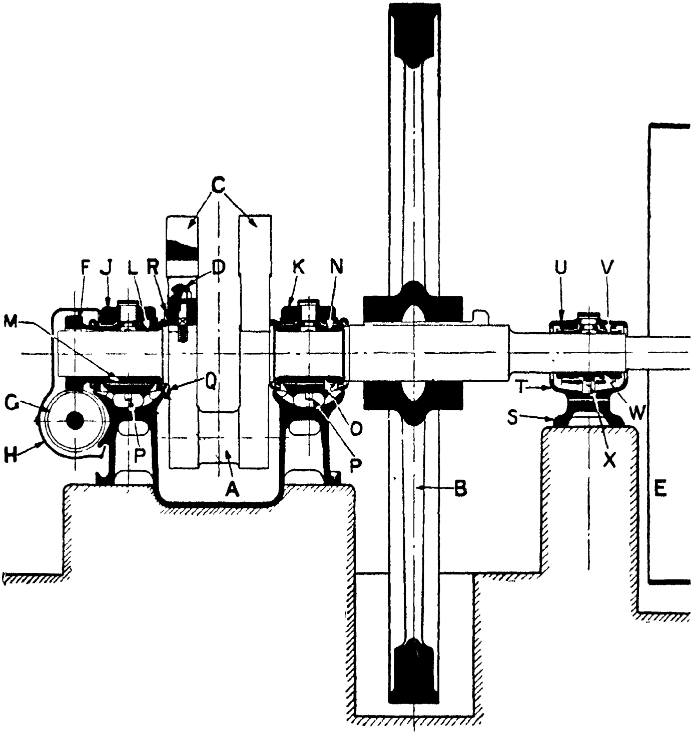


FIG. 7. CROSSLEY CRANKSHAFT, MAIN BEARINGS

A—Crankshaft.
B—Flywheel.
C—Balance weights.
D— ,, ,, studs.
E—Pulley.
F—Crank worm wheel.
G—Sideshaft wheel.
H—Wheel guard.
J—Main bearing cap sideshaft side.
K— ,, ,, ,, flywheel side.
L— ,, ,, top half, SS. side.
M— ,, ,, bottom half, SS. side.
N—Main bearing top half, flywheel side.
O— ,, ,, bottom half, flywheel side.
P— ,, ,, oiling rings.
Q—Centrifugal lubricator, bottom half.
R— ,, ,, top half.
S—Outend bearing soleplate.
T— ,, ,, pedestal.
U— ,, ,, cap.
V— ,, ,, top half.
W— ,, ,, bottom.
X –Oiling ring for outend bearing.

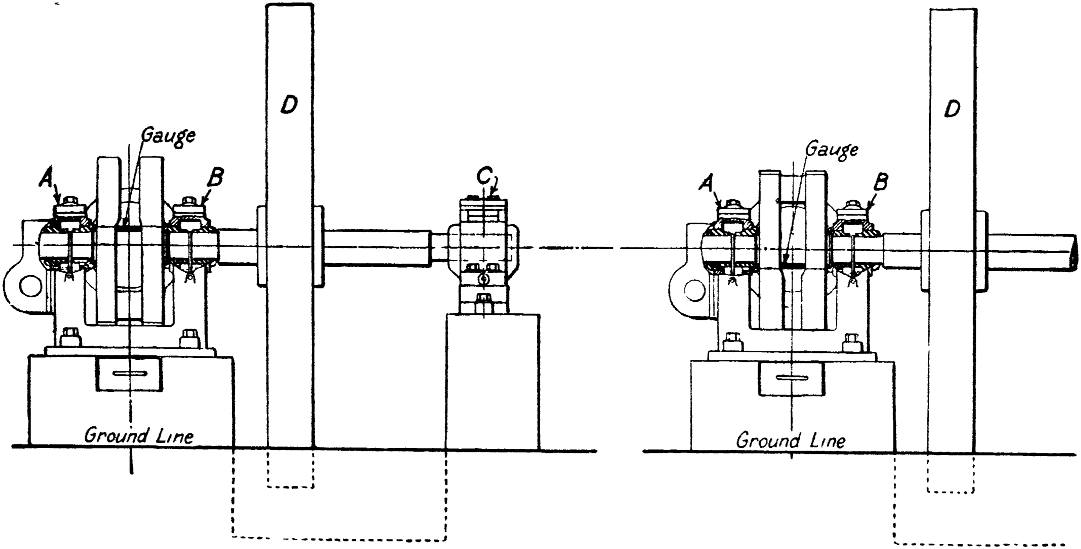
The strongest shaft is not proof against faulty alignment, and will certainly fracture if run under such conditions.

The alignment of the three bearings can be tested by measurement of the gaps between the webs of the crankshaft.

The method adopted in the case of the Blackstone engines is shown in Figs. 8 and 9, the crankshaft being shown supported in the two main bearings *A*, *B*, on the engine bed plate, and in the third independently mounted bearing *C* disposed outside the flywheel *D*. The drawings show the method of testing whether the outer bearing *C* is too high or too low. After all the bearings have been bolted down to the foundations, the crankshaft (with flywheel) mounted and the bearing caps bolted on, tests are made with a special length gauge supplied with the engine. To test vertical alignment, the engine is turned first into the position shown in Fig. 8, with the crank pin in its lowest position, and the gap between the crank webs is tested with the gauge. The engine is then turned to the position shown in Fig. 9 with the crank pin in its highest position, and the test repeated. If the gauge is slack in Fig. 8, and tight in Fig. 9, the outer bearing is clearly too low; while, conversely, if the gauge is tight in Fig. 8 and slack in Fig. 9, the outer bearing is too high.

The bearing must be adjusted up or down until the gauge slides easily in both positions. A similar test should be made with the crank pin in the two horizontal positions, to test horizontal alignment, and the bearing adjusted if necessary.

These tests should be repeated occasionally after the first erection, since the alignment may be upset by settling of the foundations. It is desirable that tests should be made at intervals of not more than six months to guard against breakdown.



FIGS. 8 AND 9. METHOD OF TESTING BLACKSTONE ENGINE BEARING ALIGNMENT

Messrs. Crossley Brothers advise measurement of the gap *X* between the crank webs with an inside micrometer, or with a dial micrometer. Fig. 10 shows, in a greatly exaggerated manner, the effect due to the outer bearing *C* being too high. With the crank pin in the upper position, the gap *X* is too great, while the gap is too small when the crank pin is in the lower position.

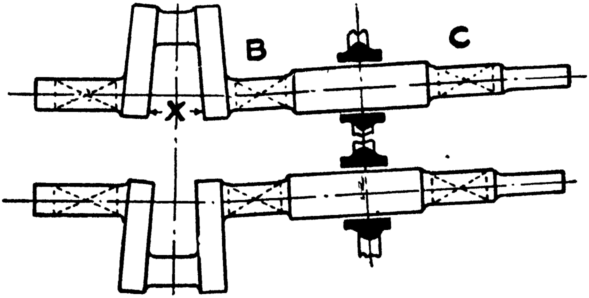


FIG. 10. CROSSLEY BEARING ALIGNMENT

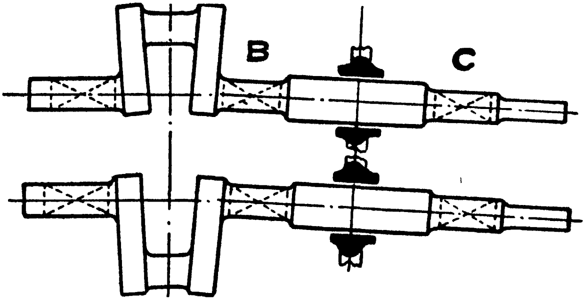


FIG. 11. CROSSLEY BEARING ALIGNMENT

Fig. 11 shows in a similar exaggerated manner the distortion of the crankshaft when the outer bearing *C* is too low.

When testing with the inside micrometer, the test points on the webs should be marked. These points should be taken in the middle of the web, and about ¼ in. from the edge.

The dial micrometer should be inserted between the webs when the crank pin is in the upper position. The adjusting nut should then be screwed up sufficiently to prevent it from moving, and the pointer of the gauge adjusted to read at zero. The crankshaft should be given half a turn and the reading of the dial noted. Similarly, readings may be taken when the crank is in its two horizontal positions to test horizontal alignment.

The greatest permissible difference between the measurements, when the crankshaft is in the two diametrically opposite positions, is from three to five thousandths of an inch according to size of engine. If these differences are exceeded, the bearings must be carefully aligned by adjustment of packing between bearing and soleplate, or otherwise according to the means provided. Frequently a bolt and nut is placed between the webs at *X* when the crankshaft is lifted for attention to the bearings.

CYLINDER LINERS

In all except the very smallest engines, the cylinder liner is made independent of the main casting or bed plate. In the Crossley engines, Figs. 2A, 12, and 13, the ends of the liner are turned a close parallel fit with the main casting. At the crankshaft end, which is comparatively cool, the joint is made watertight by an india-rubber ring let into a recess in the liner. A very little soft soap applied to the rubber ring assists when putting the liner into position. At the breech end, where the temperatures are much higher and the conditions much more severe, the joint is metal to metal with, possibly, a very thin smearing of red lead. The liner is held against endwise movement by the breech end, and by the stepped engagement with the main casting.

PISTONS AND CONNECTING RODS

Methods of inserting and removing pistons and connecting rods are shown in Figs. 12 and 13 in connection with the larger Crossley engines.

The operations carried out when withdrawing the piston and connecting rod will first be described, the reverse sequence being followed when erecting.

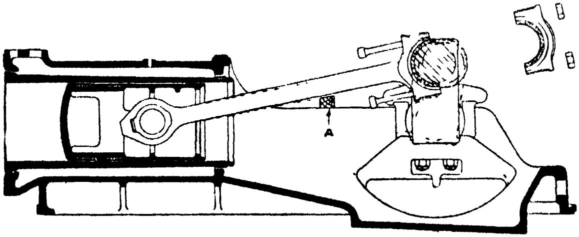


FIG. 12. DISMANTLING CROSSLEY ENGINE

The crank is turned to the vertical position with the balance weights underneath (Fig. 12), and a wooden bar *A* is rested across the sides of the bed plate to support the connecting rod after the big end is disconnected. This bar *A* should be arranged sufficiently far back to allow the balance weights to clear it when the crankshaft is turned. It should also be clear of the connecting rod by from ⅜ in. to ½ in.

The big-end bearing should now be disconnected. The cap and its lining are removed, and the bolts are tapped back as shown, so that they cannot touch and damage the journal of the crank pin when the bearing liners are removed.

The crankshaft may then be turned slightly and the back half liner rotated round the journal and withdrawn. These operations are shown in dotted lines in Fig. 12. Further rotation of the crankshaft results in the connecting rod resting on the bar *A*.

It is necessary to lower the crank pin so that the connecting rod may pass between the crank webs. The crankshaft is retained in this position by the wood packing *B* on which the balance weights rest (Fig. 13). This enables the connecting rod to be lowered into line

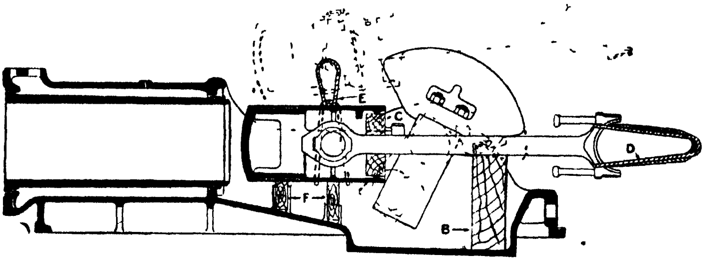


FIG. 13. CONNECTING ROD REMOVAL

with the piston, in which position it is secured firmly to the piston by wood packing *C*.

To pull the piston outwards some force may be necessary. This may conveniently be applied by a lever passed through a loop formed in a rope sling *D* passed round the big end of the connecting rod and brought forward between the balance weights. The piston should be withdrawn at first only sufficiently to expose the first piston ring.

Another rope sling *E* is passed round the piston and is secured to the overhead lifting gear, the direction of the pull being slightly forward so as to assist in the withdrawal, the piston being then pulled outwards until further movement is prevented by the crank or weights. If, in this position, the weights prevent vertical lifting of the parts, the packing *B* should be removed, and the crankshaft turned until the weights occupy the positions shown in dotted lines in Fig. 13, after which the piston may be lifted clear. On the larger engines, ribs are provided on each side of the bed to receive pieces of timber *F* on which the piston may rest when it is withdrawn from the liner.

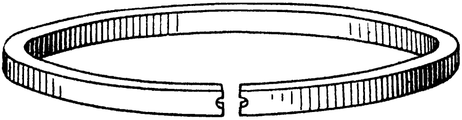


FIG. 14. PISTON RING SHOWING PIN SLOT

The pistons of gas engines, like all other internal combustion engines, are slightly tapered, being smaller at the back end where the hot gases raise the temperature considerably. Over the forward part, the diameter of the piston is approximately constant, and measures about ·0012 in. less than the liner per inch of diameter. At the back part, the piston tapers down until it is about ·01 in. less per inch of diameter. These are, of course, the differences when the parts are cold.

The piston rings (Fig. 14) are held against rotation by pins fixed in the grooves, the pins and the ends of the ring fitting one another without gripping when all the parts are in position. The pins for adjacent rings are spaced about one-third or 120 degrees round the piston.

When removing the rings or putting them in position, they should be sprung over four strips of thin sheet metal arranged longitudinally, as described in connection with petrol engines. Excessive force must not be used as the rings are easily damaged, and will not then fit the cylinder or piston.

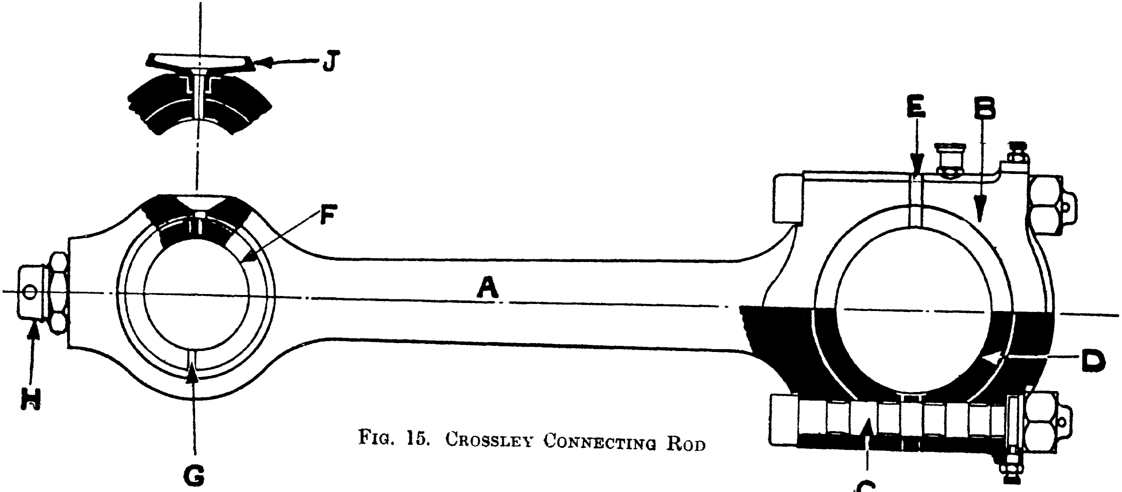


FIG. 15. CROSSLEY CONNECTING ROD

A—Connecting rod body.
B—Large end cap.
C—Connecting rod bolts.
D—Large end bearings.
E— „ „ liner.

F—Small end bearings.
G— „ „ liner.
H— „ „ adjusting screw.
J— „ „ oil trough.

The big end of the connecting rod is of the marine type as distinct from the locomotive type. Fig. 15 shows a Crossley connecting rod, part of which is in section. The bearings or brasses *D* are of bronze, and generally have whitemetal facings about ¼ in. thick to give a sufficiently yieldable body. The bearings are bored out carefully, so that their axis is exactly perpendicular to the centre line of the rod, and are then fitted to the crank-pin journal by hand scraping, using a colour indicator for the high parts. Proper adjustment of the bearings at all times is essential. They should be quite free to prevent heating, but must not, on the other hand, have enough slack to allow any knocking under the impulses of the working strokes. During fitting, the bearings are scraped to make them slacker, and the liners or packing pieces *E* are filed to make the bearings tighter. Alternatively, the ends of the bearings may be filed. The bearings must fit the journals properly when the nuts or the bolts *C* are screwed up tightly. A bearing which has been overheated will often be found to be sprung inwards, and when re-fitting, the outside of the bearing must be bedded first.

The nuts must not on any account be slacked back to give working clearance to the bearings. When tightened up, the nuts are locked by small set-screws each with its lock-nut and by split pins. As the clearance is taken up, one or more thin washers must be placed between the faces of the nuts and the split pins.

No engine should be allowed to run with a perceptible knock due to slack bearings, as a great additional load is thereby put on the big-end bolts, breakage of which may lead to a serious and expensive if not dangerous accident. New bolts should be fitted if the bearings have been allowed to knock, or if there has been any seizure of the piston, or in any case after

10,000 hours' running or, say, three or four years' use. The bolts must be tightened up equally so as to distribute the loads properly, and care must be taken to ensure that they are not twisted or lengthened. If kept bright, a crack can be detected at an early stage. To facilitate examination, Crossley big-end bolts (Fig. 16) are marked along their length with a line, and with a centre punch mark *B*. A straight edge may be placed along the line to detect twist, or the bolt may be mounted in V blocks and a scribing block used for the same purpose. The makers supply with their engines particulars of the maximum permissible elongation and twist. For example, on a bolt having the dimension *A* equal to 9 in., the maximum extension permissible is ·018 in., and the maximum twist is $\frac{1}{16}$ in. When the bolt has been deformed to such an extent that either of these figures has been exceeded, new bolts must be fitted.

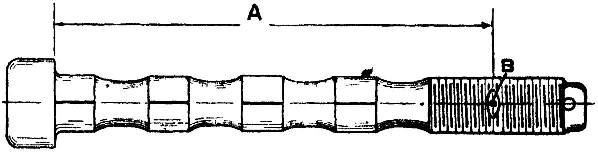


FIG. 16. CROSSLEY BIG-END BOLT

Connecting-rod bolts, if of nickel or other special steel, should not be annealed. Such bolts are made from specially heat-treated steel to obtain the best combination of strength and toughness. When replacements are necessary, new bolts should in general be obtained from the makers. If, however, the bolts are made of Yorkshire iron or mild steel, they should be annealed about once a year to reduce risk of crystallization and consequent breakage.

The small-end bearings are of brass, and are sometimes divided like the big-end bearings, but in the case of the Crossley engines, Fig. 15, the liners are placed in a ring-shaped end to the connecting rod. As the amount of movement in these bearings is small, and the chances of heating correspondingly less than in the case of big-end bearings, whitemetal is rarely employed. The gudgeon pin is held against rotation in the piston and the bearings are fitted to it, as in the case of the big end, with freedom but without slack. The bearing surfaces are scraped to slacken, and the abutting ends of the bearings or brasses are filed to tighten the bearings. Alternatively, the liner *G* may be filed. The bearings are tightened up firmly by the adjusting screw *H* provided with a lock-nut.

BEARINGS

The crankshaft or other bearings are fitted with liners or brasses of bronze, or of bronze lined with whitemetal. These bearings are all fitted and finished substantially the same as the big-end bearings referred to in the preceding section. It is therefore unnecessary to do more here than to emphasize the necessity of careful fitting to obtain a large bearing area and accurate adjustment to avoid either knock or tightness. Lubrication will be dealt with in the lubrication section.

FLYWHEELS

The flywheels of gas engines, and particularly those with one cylinder only, are very heavy to ensure uniform speed of rotation. The flywheel must run true and be absolutely rigid with the crankshaft. Further, since flywheels are always made of cast iron, care must be taken in fitting to the crankshaft to avoid strain or distortion.

Flywheels are of three types—

1. Solid, generally with curved arms to avoid strains due to unequal cooling of the different parts of the casting.
2. Solid rim and split boss.
3. Split rim and boss.

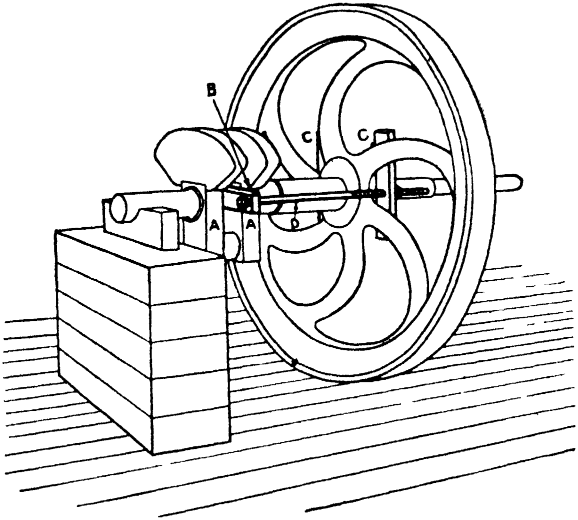


FIG. 17. FLYWHEEL PULLING GEAR

In the solid type the boss is bored accurately to gauge, and the portion of the crankshaft on which the flywheel fits is accurately machined to obtain a tight fit between the two.

The flywheel is drawn on to or off the shaft by a screw pulling gear, the arrangement recommended by Crossley Brothers being shown in Fig. 17. The flywheel is first placed upright on the floor or preferably is held by overhead lifting gear, and the shaft is entered as far as possible. To ensure correct relative position, the key or keys should be entered at the first opportunity. The end of the crankshaft should be supported, as for instance, by a Vee block and packing case, bench or other support. The pulling gear is then rigged up as follows: A plate is arranged across the near crank web *A*, and is connected by two long bolts *D* to two plates *C*, each arranged across two arms of the flywheel.

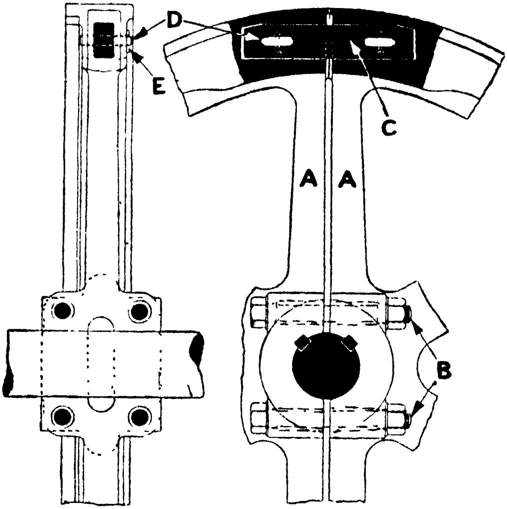


FIG. 18. FLYWHEEL WITH SPLIT RIM AND BOSS

The bolts must be screwed for a considerable part of their length. The nuts must be tightened up uniformly, putting an equal tension on the two bolts as far as possible until the flywheel is drawn on to its correct position. Judicious "bumping" on the boss by means of a heavy piece of wood, alternating with the tightening of the pulling gear will help considerably. The keys should then be driven in tightly.

When removing the flywheel the operation is similar, but the bar is placed across the end of the crankshaft remote from the crank, and the plates are arranged across the flywheel arms on the side near the crank.

In the case of flywheels having a divided boss, the boss is bored out to fit the shaft accurately, so that it may be clamped tightly on to the shaft without straining the casting. The flywheel and shaft may be held as described previously, but it is necessary first to slacken the bolts holding the parts of the boss together, and to separate slightly the two parts of the boss by driving iron wedges between them. This enables the shaft to be entered easily without any pulling gear. The keys may then be entered to position the flywheel, the boss bolts tightened up, and the keys driven in tightly.

Fig. 18 shows a large Crossley flywheel in two separate halves, as fitted to the larger single-cylinder engines and to all double-cylinder engines.

The meeting surfaces of the rim must be scrubbed clean with paraffin, to ensure close metal to metal contact before final assembly.

One of the two halves is lifted into the flywheel pit and is packed up to its correct level, and carefully arranged in position to suit the crankshaft, which is then lifted up and lowered into its bearings. The dowels *C* are then placed in position in the holes in the lower half. The upper half of the flywheel is then lifted and lowered carefully into position, and the wedge-shaped cotters *D* inserted and tapped lightly in. The keys are next inserted into the keyway an inch or two, that is, only sufficiently to ensure that the flywheel is in its correct angular position. The boss is next clamped firmly on to the shaft by the boss bolts *B*, and the cotters may then be driven home finally and secured against slacking back by split pins *E*. The flywheel keys are then driven right in.

KEYS AND KEYWAYS

All keys and flywheel keys in particular must be driven up tight, and must be kept tight. Never try to drive a new key home without fitting it little by little and making sure it is not too large or too large a taper. A slack key will sooner or later certainly cause breakage of the flywheel or the crankshaft or both. It is impossible to pack up a slack key with liners. A new key must be fitted. Keyways are generally machined parallel, and care must be taken that they are not burred or otherwise rendered smaller at the entering end. This defect is shown in Fig. 19 in an exaggerated manner. The key *A* is slack for its whole length, except at the entering point. The keyway may be made slightly taper to ensure tightness when the key is driven in.

The key must fit at the sides and at the top and bottom, as shown in Fig. 20. No slackness whatever is permissible, except at the parts indicated in dotted lines. A key may be filed slightly hollow at this part when a keyway has been damaged, for instance, by a slack key, but it is better to true up the keyway. If there is any slackness whatever, so that a key starts rocking or working, it will ultimately burst the boss or do other serious damage. Fig. 21 shows a key which bears only at the middle of the top and bottom keyways, and will ultimately cause a breakdown. The figure shows this defect exaggerated, but it must be clearly understood that the smallest clearance or slackness towards the corners will cause trouble. Fig. 22 shows a loose key taking a one-way drive.

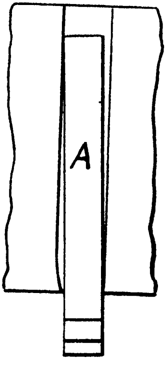


FIG. 19
SLACK KEY

With a new engine, keyways and keys are generally machined carefully to size, so that there is very little fitting, but in carrying out repair work hand fitting may be necessary. To fit keys properly, practice is essential, but some guidance may be furnished. Careful attention must first be given, as suggested above, to the keyways. The key must be of hard steel as its corners are subjected to severe compressive forces, and they are not well backed up by a mass of metal. Tough axle or similar steel is satisfactory. For small keys, an old file which has been annealed is quite suitable. The key is fitted by a succession of trial entries. For each trial it is driven in lightly and then removed, a head as shown in Fig. 23 being useful for this purpose.

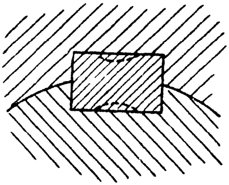


FIG. 20. PROPERLY FITTING KEY

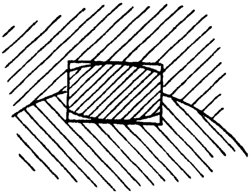


FIG. 21. BADLY FITTED ROUNDED KEY

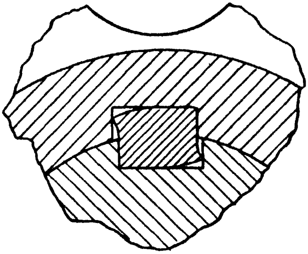


FIG. 22. SLACK KEY

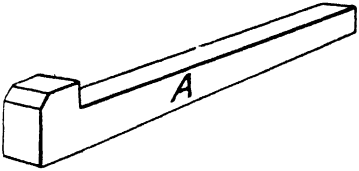


FIG. 23. SHAFT KEY

Material is then filed off in accordance with the markings to ensure proper engagement along the whole length. One or two inches, according to the size of the key, should be left for the final drive home. The key and keyways should be covered with thick gear oil, which will help to prevent seizing of the metal and will facilitate removal. If a new key has been fitted because of slackness, it is as well to see to the balance weights; the shock may have caused slackness there also.

BREECH ENDS

The breech end carries the inlet and exhaust valves and the ignition block, while the interior forms the combustion chamber. Fig. 24 shows a sectional transverse view, and Fig. 25 a side view of the Crossley breech end. The joint between the front face of the breech end and the ends of the main casting and cylinder liner is made with thin sheet asbestos. The Campbell Gas Engine Co. recommend for their engines best sheet asbestos soaked in boiled linseed oil and smeared with blacklead. In general, as thin jointing as possible should be used, provided this does not unduly affect the compression ratio.

The nuts on the studs securing the breech end in position must be screwed up carefully to avoid distortion and unevenness of the joint. All the nuts should be gone over a number of times, tightening each up only a little at a time, say one-quarter to one-half a turn, so that all are gradually tightened up uniformly. It is advisable also to tighten up opposite nuts in succession; thus, first a top nut, then a bottom nut, next a side nut, and then an opposite side nut, and so on.

These remarks on the bolting up of joints apply also to the several other smaller joints on the breech end,

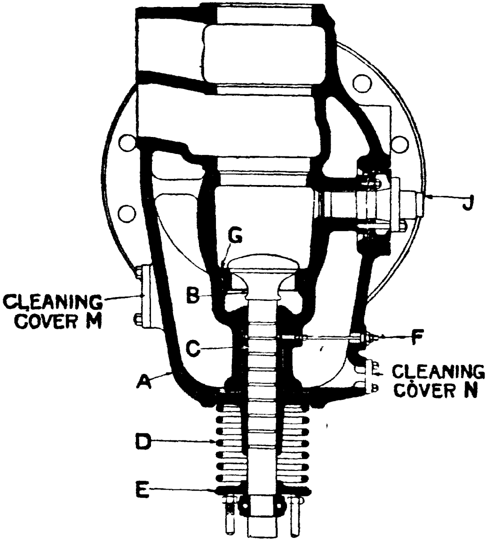


Fig. 24

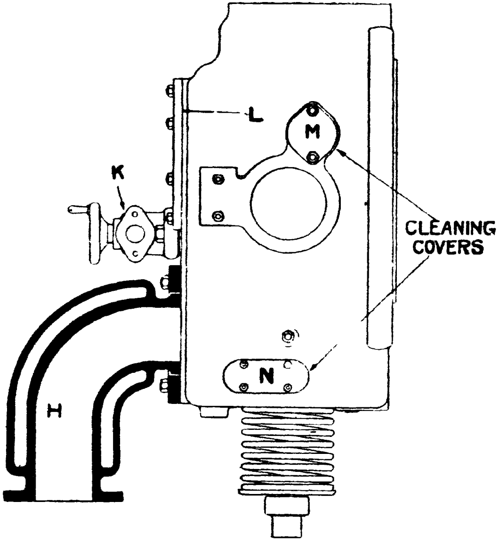


Fig. 25

Crossley Breech End

A—Breech end.
B—Exhaust valve.
C— ,, ,, guide.
D— ,, ,, spring.
E— ,, ,, ,, plate.

F—Exhaust valve oil pipe.
G— ,, ,, seating (not used).
H—Exhaust bend (water cooled).
J—Igniter block.

K—Starter valve.
L—Breech end cover.
M—Cleaning cover.
N— ,, ,,

which have to be made from time to time. The copper-asbestos jointings frequently used will be unevenly compressed if tightened hard at one side before the other is held, and will not seal the joint properly.

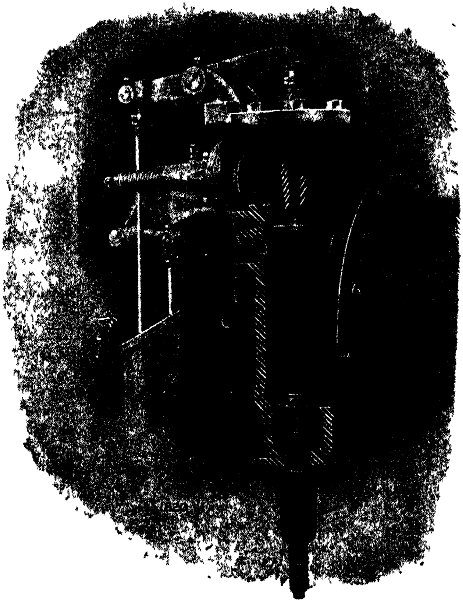


FIG. 26. HORNSBY-STOCKPORT BREECH END

Fig. 26 shows a sectional view of the Hornsby-Stockport breech end with valves.

VALVES

The exhaust valve is generally arranged at the lower side of the breech end, and opens upwards while the inlet valve is vertically above and opens downwards.

The Crossley exhaust valve is shown in section in Fig. 24, and also in Figs. 2, 2A and 27. In this construction, the exhaust valve, which is generally of cast iron, seats directly in the breech end or in a detachable

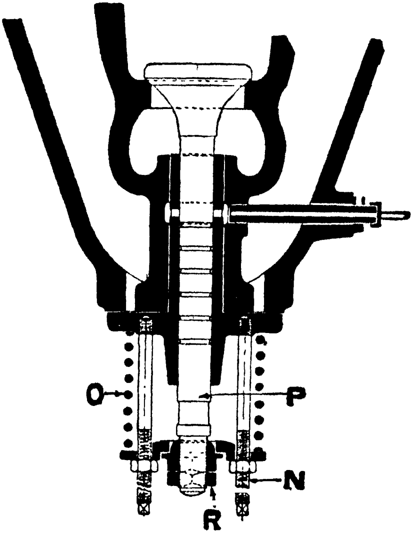


FIG. 27. CROSSLEY EXHAUST VALVE

cat *G*, and is guided by a detachable guide *C*. The guide is bolted up in position, and the valve lowered into it from above through the opening in which the inlet valve cage is located when in position. The valve is held by a T-ended grinding spindle which is screwed into a hole in its upper surface. To compress the valve spring, the attachment shown in Fig. 27 may be used. The spring *O* and plate are placed in position, and the spring compressed by nuts on studs *N* which are screwed temporarily into holes on the underside of the breech end or the flange of the guide. When the spring is sufficiently compressed, the nuts *R* are screwed on to the valve stem, after which the studs *N* may be removed. The nuts *R* must be screwed up until the pull of the spring on the valve prevents lifting when running without load, except, of course, when operated by the cam gearing.

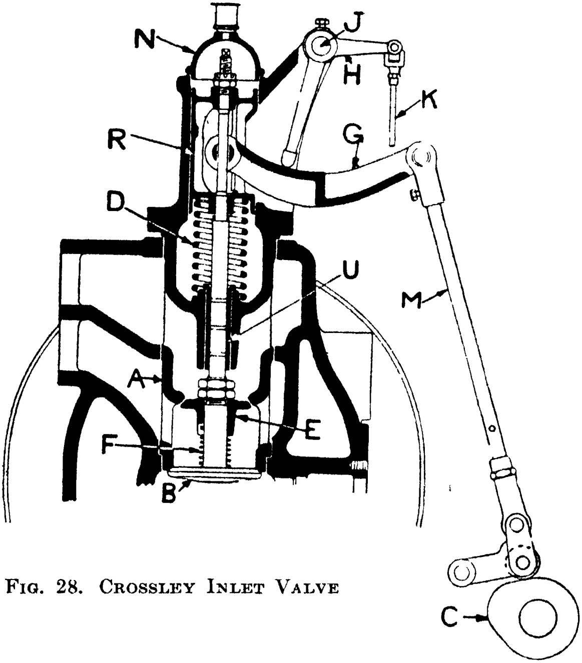


FIG. 28. CROSSLEY INLET VALVE

The Crossley inlet valve *B*, Fig. 28, is carried with its spring *D* in a detachable block *A*. The gas valve *E* is slidable on the main valve shown. It is pressed upwards by a spring *F* against nuts which must be so adjusted that the gas valve opens slightly after and closes slightly before the valve *B* to prevent leakage of gas into the air suction silencer. Both valves are drawn upwards on to their seats by a spring *D* acting on a guide *R*. The valve is opened downwards against the spring *D* by a cam *C*, rod *M*, and rocking lever *G*. The fulcrum or pivoting point of the lever *G* consists of the end of the approximately vertical arm of the belt-crank lever *H*. The governor, through a link *K*, controls the position of the fulcrum, and hence varies the lift of both the gas valve *E* and the mixture inlet valve *B*.

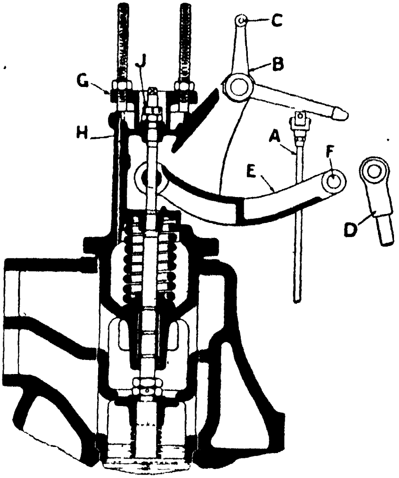


FIG. 29. COMPRESSING INLET VALVE SPRING

The valve spring *D* is compressed when assembling, before the cover *N* is placed in position, by the attachment shown in Fig. 29, and consisting of a flange *C*, and two studs adapted to be screwed temporarily into the head of the admission valve block. The flange is screwed down, forcing the valve guide *H* down, and compressing the spring until the nuts *J* can be screwed on to the valve stem.

The valves are ground in by oiling the faces and sprinkling emery powder thereon. Carborundum should not be used for this purpose. The T-headed grinding spindle referred to above is then screwed into the valve and locked by a nut. The valve is pressed on to its seat lightly and twisted to and fro, being lifted at intervals, and turned to a different position. Next remove the valve and wipe it and its seating quite clean. Then replace it, and work it round lightly on its seat while dry. The valve and seating should, if properly ground in, show a continuous engaging surface all round. If this result has not been obtained, the grinding should be repeated as many times as may be necessary.

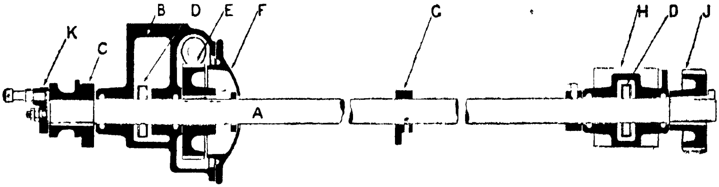


FIG. 30. SIDE OR CAM SHAFT

A—Side shaft.
B— „ bracket, cylinder end.
C—Combined cam.
D—Oiling ring.
E—Governor wheel.
F—Cover for side shaft bracket.
G—Lubricator cam.
H—Side shaft bracket crank end.
J— „ wheel.
K—Driving disc for electric ignition.

VALVE OPERATION

The usual side shaft shown in the Crossley general arrangement, Fig. 1, and also in Fig. 30, is mounted in two bearings, one at each end. The shaft is arranged below the level of the crankshaft, and is driven by skew or worm gearing, the driven wheel *J* being keyed on the overhung front end of the side shaft. The lower gear wheel dips into an oil bath, while the adjacent bearing is lubricated by the usual ring oiler, in which a ring substantially larger than the shaft runs thereon and dips into an oil bath.

A somewhat similar example of this class of driving mechanism as applied to a Tangye gas engine is shown in Fig. 31.

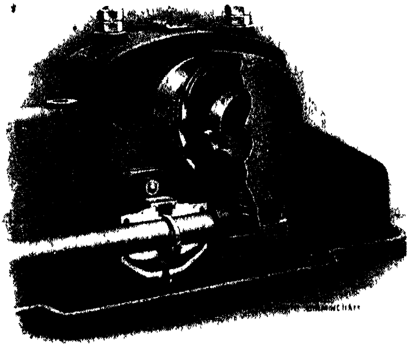


FIG. 31. TANGYE CAMSHAFT DRIVING GEAR

Referring again to the Crossley side shaft, Fig. 30, a gear wheel *E* keyed to the shaft drives the governor spindle, and a combined cam *C* operates both the inlet and exhaust valves.

When erecting, care must be taken that the worm gearing for the side shaft is properly engaged. A tooth on one of the wheels is invariably marked to engage the gap, which is also marked between two teeth on the other wheel. In general, each wheel is marked *O*, and the marks on the two wheels must coincide when the two are rotated to a suitable position. The markings used on Crossley engines are shown in Fig. 32.

Owing to the speed at which gas engines run, and the inertia or lag of the incoming mixture and the exhaust gases, the valves are set to open considerably before and close after the dead centre positions. The engines of different makers differ somewhat as regards timing, owing to variations in design

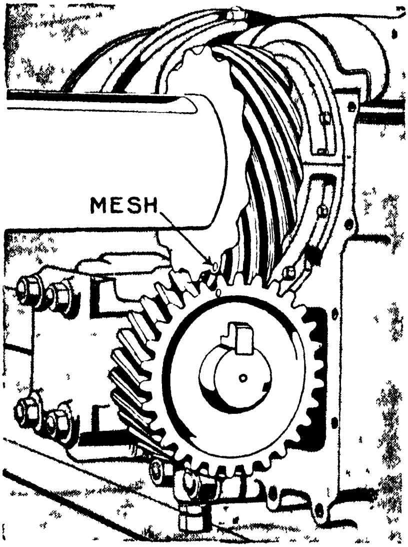


FIG. 32. CAMSHAFT DRIVING GEAR

It may be necessary on new or on repair work to check or to reset the valve timing, and the following will explain how this is done.

Fig. 33 shows diagrammatically the positions of the crank pin when the different events of the cycle take place, the timing shown being that adopted on Crossley engines.

Consider as the starting point of the cycle the position of the flywheel when the crank pin is at *A* on its

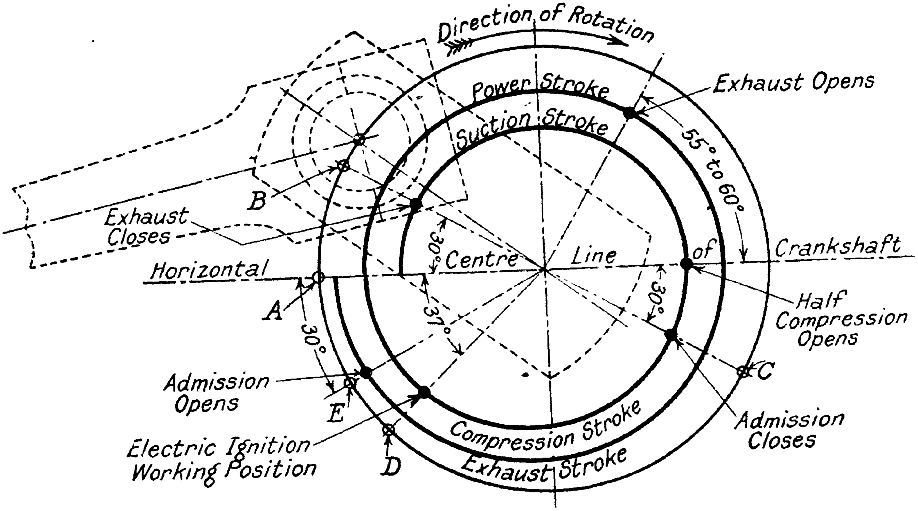


FIG. 33. CROSSLEY TIMING DIAGRAM

dead centre at the beginning of the suction stroke. With the ignition switched off, and the gas and starting valves closed, turn the flywheel through 30 degrees to *B*, when the exhaust valve should just have closed and the exhaust roller should only just be leaving the exhaust cam. Then turn the flywheel to *C*, where the inlet valve should similarly just be closing.

Next turn the flywheel to *D*, when the magneto trip lever should be released. Further rotation should show the exhaust valve opening at a point on the working stroke when the crank is 55 to 60 degrees ahead of the outer dead centre. The admission valve next opens when the crank is at *E*, 30 degrees ahead of the inner dead centre. From *E* to *B* there is an overlap period, when both air inlet and exhaust valves are open. It should, however, be remembered that the gas valve opens after the main admission valve, so that only air enters at first.

The admission valve will only open and close at the correct times if the clearance between the radius lever *G*, Fig. 28, and the end of the upright arm of the fulcrum lever *H* is maintained between $\frac{1}{64}$ in. and $\frac{1}{32}$ in. for all positions along the radius lever. The upper limit given should not be exceeded and, on the other hand, the fulcrum lever must never be allowed to stick, as the governor would then be out of action, and the speed might increase to a dangerous extent.

The clearance between the exhaust cam and the exhaust roller should be kept at $\frac{1}{32}$ in. to ensure correct timing of the exhaust valve. It may be necessary to stone the cams owing to excessive wear. This will affect the valve timing and it may be necessary to fit new cams.

LUBRICATION

Modern machinery depends for its successful operation upon lubrication. Metal surfaces cannot work together and carry the necessary heavy loads, unless they are separated by a thin film of oil having the necessary body to prevent it from being squeezed out under the conditions of load, speed, and temperature to which the bearing or other part may be subjected.

The cylinder walls and the stem of the exhaust valve work under the most severe conditions, and provision generally is made to supply them with oil continually from a reservoir by means of pumps. In larger engines, the big-end bearing and the small-end bearing or gudgeon pin are similarly supplied with oil. A number of small plunger pumps, one for each of the parts in question, are worked from the camshaft and deliver oil through sight feed glasses, the flow to each part being individually adjustable.

The Crossley central mechanical lubricator comprises four pumps supplying oil respectively to the cylinder, exhaust valve, big-end bearing, and gudgeon pin, one of the pumps being shown in section in Fig. 34.

The filling opening to the container *A* contains a wire gauze filter, which must be examined for soundness when assembling. The spindle *K* enters the container through a bearing which must be carefully packed, and is oscillated by an arm *F* and an operating rod *G*, worked from a cam or eccentric at about the middle of the length of the camshaft. To the shaft *K* are secured levers *J*, one for each pump. Each plunger *B* must fit its cylinder without any perceptible slack. An extension of the plunger is fitted with a finger ring, by means of which each pump may be worked independently. The extension also carries a sleeve *N*, which acts as a stop and so limits the effective stroke. The gland *X* for the extension requires special attention, as it has to stand the full delivery pressure of the oil.

The suction and delivery ball valves may be tapped lightly through a brass rod on to their seats, which

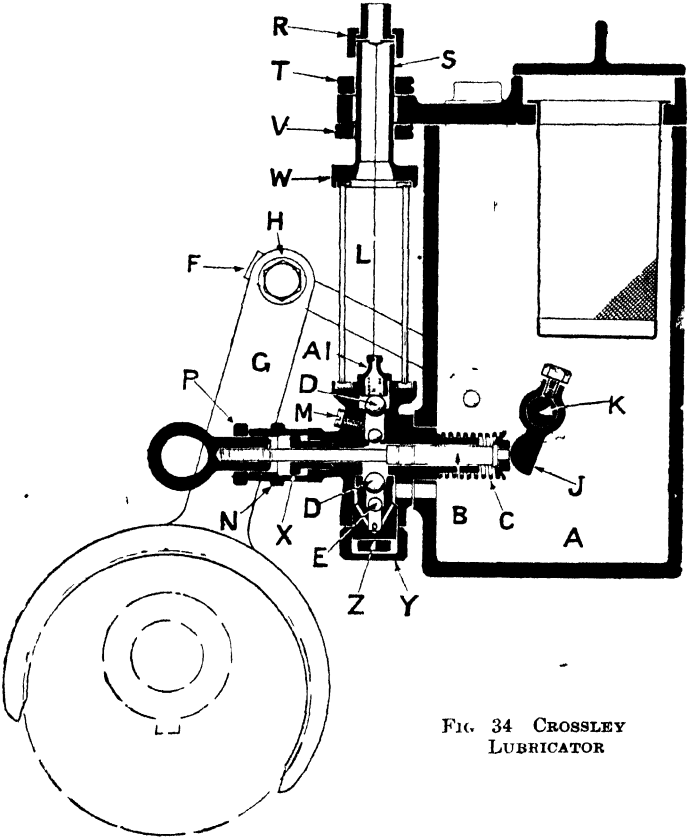


FIG. 34 CROSSLEY LUBRICATOR

A—Container
B—Plunger
C— ,, spring
D—Ball valve, large
E— ,, ,, small.
F—Driving arm
G—Operating rod
H—Connecting rod pin
J—Internal tappet lever.
K— ,, ,, ,, layshaft
L—Sight-feed glasses containing salt water.

M—Vent screw
N—Adjusting sleeve
P—Lock nut for adjusting sleeve
R—Union nuts for oil pipes.
S—Shank connections to feed glasses
T—Lock-nut for shank *S*
V— ,, ,, ,, *S*
W—Joint washers for glasses.
X—Gland ring for plunger
Y—Cap for suction valve cage.
Z—Cage for suction valves.
*A*1—Delivery nozzle.

are of much softer material so as to make them fit properly. The sight-feed glasses *L* should contain a saturated solution of salt, and must be clamped firmly endwise, care being taken that the leather washers at the ends are sound.

The small copper pipes which lead the oil to the several points where it is required must be filled with oil before starting. Before the joints at the ends remote from the pumps are tightened up, the pumps should be worked by hand until oil appears at these ends. All the pumps may be worked together by the lever *F* or one only at a time by its finger ring without the driving gear being disconnected.

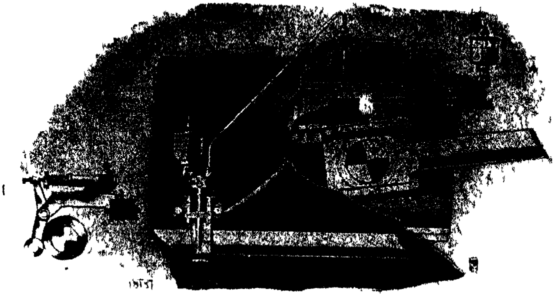


FIG. 35. HORNSBY-STOCKPORT LUBRICATION OF CYLINDER, PISTON, AND GUDGEON PIN

Fig. 35 shows, in the case of a small Hornsby-Stockport engine, how the oil is supplied to the cylinder walls and piston, and also to the gudgeon pin bearing. In the latter, oil from a drip-feed reservoir or from a pump is supplied steadily to a wiper, from which it is conveyed by a conduit inside the piston to a point above the gudgeon bearing and drops thereon as shown. The parts of the wiper must touch only lightly or, preferably, not at all, provided that they are sufficiently near to transfer the oil drop by drop.

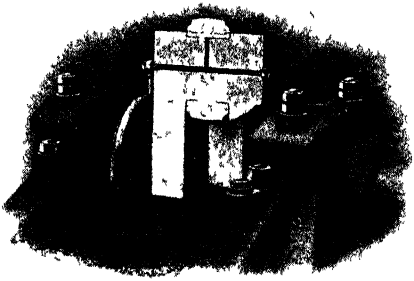


FIG 36. TANGYE CENTRIFUGAL OILER FOR CRANK PIN (BANJO TYPE)

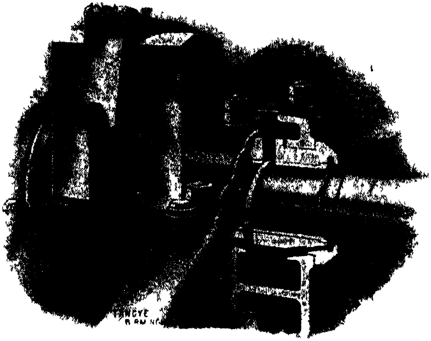


FIG. 37. TANGYE RING OILERS FOR CRANKSHAFT MAIN BEARINGS

The oil connection *F* to the stem of an exhaust valve is shown in Fig. 24.

Fig. 36 shows a centrifugal oiler for the crank pin of a Tangye engine. The oil supplied drips into a ring which turns with the crankshaft, and is concave or hollow on the inside. The oil, under the action of centrifugal force, flows outwards to the crank-pin bearing through the pipe and drilled oilways shown.

For the various other parts of an engine, ring oilers, drip lubricators, or merely oil holes or nipples are provided.

Figs. 37 and 31 show ring oilers applied to Tangye engines in connection, respectively, with a crankshaft bearing and a camshaft bearing adjacent to the crankshaft. The rings must be large enough to dip well down into the oil reservoir, and must be quite free and loose. Provision for the return of surplus oil to the reservoir will be noted.

The several copper pipes must be laid neatly in position and clipped as required, sharp bends being avoided.

Considerable care is necessary in the final assembly, and all the lubrication system, especially exposed surfaces, must be kept scrupulously clean. If the engine is in a dusty atmosphere, frequent cleansing of the whole system thoroughly will pay for itself.

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